PRINCIPLES *of*

REFRIGERATION

A COMPREHENSIVE TREATISE ON FUNDAMENTAL PRINCIPLES OF OPERATION OF ICE MAKING AND REFRIGERATING MACHINERY, PROPERTIES AND VALUES OF PRINCIPAL MEDIA USED IN MODERN REFRIGERATING APPARATUS; TRANSMISSION OF HEAT, FUNCTIONS AND VALUES OF INSULATING MATERIALS; CONSTRUCTION AND OPERATION OF VARIOUS PARTS OF REFRIGERATING APPARATUS AND APPLICATION OF REFRIGERATION TO ITS VARIED USES

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PREFACE

The object of this work has been the presentation of the elementary and fundamental principles underlying the operation of ice making and refrigerating machinery; the properties and values of principal media used in modern refrigerating apparatus; the fundamental laws underlying the transfer of heat; the design, construction, and operation of the various parts of refrigerating apparatus; together with the application of refrigeration to some of its most important uses.

The book will be found useful for beginners in the study of refrigeration, and as a textbook for students in technical schools. It provides information and data for the practicing refrigeration engineer. It is believed that the work will prove of special interest to erecting and operating refrigerating engineers, refrigerating machinery sales engineers, persons employed in building refrigerating machinery, and those employed in ice making, cold storage plants, or other establishments equipped with refrigerating machinery. In general, the work contains information that it is necessary the refrigerating engineer should know in order that he may have complete and up-to-date knowledge of the theory and practice in this field of endeavor.

The work is in everyday language, and as free as possible from higher mathematics. The method of treatment has been to present a comprehensive treatise on the fundamental principles. With a firm grounding of these fundamental principles, the practitioner is enabled to intelligently design or operate refrigerating machinery. The theoretical and fundamental operating principles are given attention first. This is followed by numerous practical considerations and the application of the fundamental principles to the economic production of ice and refrigeration for various purposes.

This book is used by the National Association of Practical Refrigerating Engineers for their National Lecture Course. It

is used in this form as a method of study for the practical operating refrigerating engineers belonging to the Chapters of that Association located in various cities of the United States, as well as by its members-at-large.

The author has drawn extensively on his wide experience as a refrigeration engineer and teacher of refrigeration engineering for materials for this work. During his practice with manufacturers of ice making and refrigerating machinery, and his instructional work in the Ohio Mechanics Institute, Cincinnati, O.; the University Extension Division of the University of Wisconsin, Madison, Wis.; the Siebel Institute of Technology, Chicago, Ill.; and at the Chicago Chapter of the National Association of Practical Refrigerating Engineers, many notes were collected which were used directly in the compilation of this book.

The author has also obtained considerable information from technical journals, manufacturers' catalogs, refrigerating engineers, and trade associations, for which proper credit has been given. The author also wishes to gratefully acknowledge here the assistance given to him by various associations, publishers, manufacturers, engineers, and others, during the compilation of the subject matter of this work.

The author is especially grateful to E. S. Libby, Professor of Refrigeration, Armour Institute of Technology, Chicago, Ill.; H. G. Venemann, Professor of Refrigeration, Purdue University, Lafayette, Ind.; Chairman and member, respectively, of the Educational and Examining Board, N. A. P. R. E.; Fred I. McCandlish, Past-President of the National Association of Practical Refrigerating Engineers; and others, for valuable assistance received during the compilation of the subject matter of this work.

W. H. Motz.

CHAPTER I.

INTRODUCTION AND ELEMENTARY REFRIGERATION SYSTEMS.

Development of the Science of Refrigeration.—The practice of cooling bodies below the temperature of the atmosphere has been followed for centuries. One of the early methods consisted of the evaporation of a part of the liquid to be cooled by putting the liquid into porous vessels which were hung in a current of cool moving air. This practice was followed in localities where the atmosphere was warm and dry. Another early method consisted of the construction of artificial caves or cellars in the ground, into which perishable goods were placed to retard decomposition. In most all countries a temperature of 50° to 60° F. may be obtained in these cellars. Another early method of obtaining low temperatures consisted of the use of freezing mixtures. Such mixtures as water and saltpetre, snow or ice and saltpetre, snow and salt, etc., have been used for ages. In a like manner use was made of ice. The ice, after being harvested in the winter, was stored in caves in the ground. In this manner man was able to secure a supply of ice to preserve his perishable foods during the summer months.

Thus from the creation of man until nearly modern times the only available means of producing refrigeration were those mentioned above. It was not until the year 1755 A. D. that the first experiments were performed in order to discover a means of producing refrigeration *mechanically*. Therefore mechanical refrigeration may be said to date from the year 1755, at which time the temperature-pressure relations of certain refrigerating fluids were observed.

During the next seventy-five years many experiments were performed and many experimental machines were constructed. The vacuum machine, sulphuric acid machine, water machine, etc., were brought out during this period.

However, the real foundation for the development of the compression and absorption machines was made in the year of 1823, when it was discovered that certain refrigerating fluids could be liquefied, after

being compressed to a high pressure and then cooled. These experiments were performed by Michael Faraday, of England.

Then in 1834 the first compression machine was invented. This was a crude ether compression machine but was the first machine to produce refrigeration or ice in commercial quantities by mechanical means. This was brought out by Jacob Perkins, an inventor and engineer, born in Massachusetts.

In the year of 1850 the production of refrigeration by means of the cold air machine was invented by John Gorrie, an American. Five years later, in 1855, the first absorption machine was invented. This was a crude affair, but it was the forerunner of the modern absorption system. This system was developed by Ferdinand Carré of France. A steam coil for distilling off the ammonia was not used until 1865, and in the same year the first transparent ice was made from distilled water in the United States.

In the years of 1873-75 the first successful ammonia compression machines were introduced by C. P. G. Linde, of Germany; and David Boyle, of the United States. From 1875 to 1890 many new forms of apparatus were produced and certain improvements were made.

Until the year 1890 the practical utilization and commercial application of refrigeration were quite limited, and the development of the art of refrigeration had seemed to come to a stand-still, but there occurred in the year 1890 an incident that awakened the general public to the possibilities of the use of mechanical refrigeration. This incident was the greatest shortage in the crop of natural ice that has ever occurred in the United States. Thus to this peculiar incident may be accredited the impetus that started the rapid development and utilization of mechanical refrigeration in the United States.

Another important happening in the same year was the conception of the first trade journal in the world devoted exclusively to the ice and refrigerating industries. Thus the premier refrigeration journal, *Ice and Refrigeration,* was conceived in 1890 and published for the first time on July 1, 1891.

As the basic principles underlying the operation of refrigeration apparatus were discovered and brought out before the year 1890, it was only necessary to develop the many improvements in design and operation in the succeeding years in order to provide means for universal application. Many improvements were incorporated into the mechanical design of the compressors. More efficient methods of condensing the discharged gases were brought out. Fittings, accessories, etc., were improved and standardized more or less. New flooded or gravity feed systems for the evaporating surfaces were introduced. The process of the manufacture of clear merchantable ice from raw water was introduced and perfected. Thus during the last forty years the ways and

means of producing refrigeration have been so improved that it can be practically applied to many processes in the industrial world in an efficient manner.

Methods of Production of Refrigeration.—Among the commercially practical refrigerating systems the following are the most important:

1. Natural and manufactured ice.
2. Compression system using liquefiable fluid (as anhydrous ammonia).
3. Absorption using liquefiable fluid (as aqua ammonia).

While the foregoing methods of production of refrigeration are the most important commonly used systems, it is probable that a more general understanding of refrigeration, as a whole, may be obtained by studying the following outline, which, in addition to giving those which are commonly used, gives the other systems which are sometimes used for more or less special applications to refrigeration.

OUTLINE OF METHODS FOR PRODUCING REFRIGERATION.

- Non-mechanical
  - Cooling by evaporation of liquids
    - water
    - ether
    - others
  - Cooling by melting of solids
    - ice, natural and manufactured.
    - others
  - Cooling by freezing mixtures
    - ice and salt
    - ice and calcium chloride
    - others
  - Cooling by sublimation
    - Carbon dioxide ice
- Mechanical
  - Direct cooling by evaporation of liquids
    - compression system
    - absorption system
    - vacuum system
    - adsorption system
  - Indirect cooling by gases
    - compressed air
    - forced air
  - Indirect cooling by liquids
    - brine circulation
    - water circulation
    - others

Refrigeration by Evaporation.—One of the early methods consisted of the evaporation of part of the liquid to be cooled in porous vessels which were hung in a current of cool moving air. The liquid which was commonly used for this purpose was water and the method of using water for cooling in this method is illustrated as shown in Fig. 1.

For the purpose of cooling food a porous earthen jar was first submerged in water for a few hours so that the water would penetrate into the walls of the jar. After this the jar was removed from the water

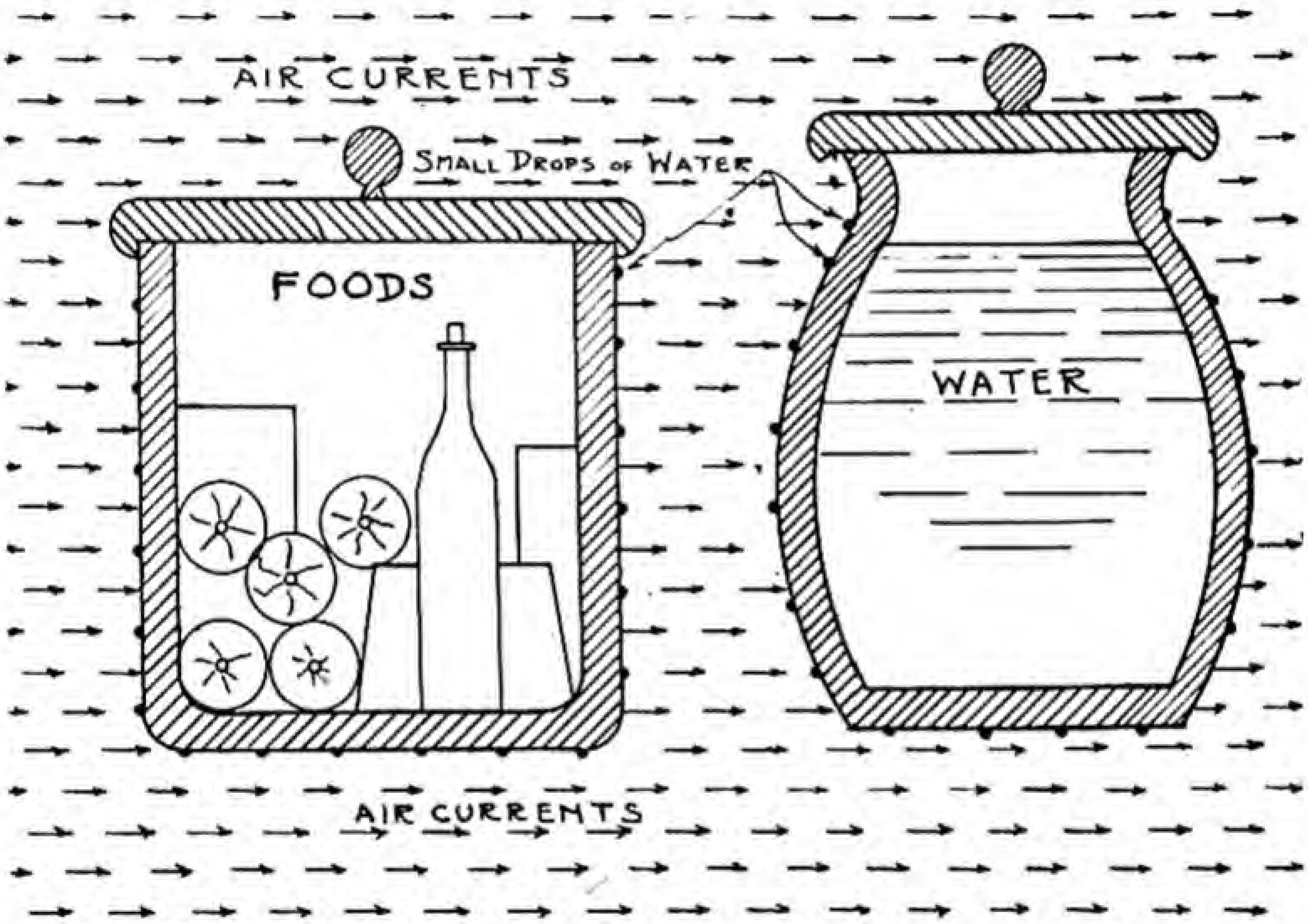


Fig. 1.—Cooling by Evaporation.

and the food products to be cooled stored in jars. The jar was then placed into a cool current of moving air, as shown in Fig. 1. The evaporation of the water contained in the porous walls of the jar was sufficient to lower the temperature of the material inside of the jar a few degrees below that of the surrounding atmosphere. The water in the porous walls was in the liquid form, and in order to change the water from a liquid form to the vapor form, or, in other words, to evaporate it, it is necessary that a certain amount of heat be added to the water. This heat, which is absorbed by the water in evaporating, comes from the material stored in the jar, thereby producing the required cooling effect. As shown in Fig. 1, water may be placed in an earthen jar and cooled in the same manner as foods. In case of cooling the water in the earthen jar, a small amount of water passes through the wall and

evaporates at the exterior surface of the wall, thereby producing the desired cooling effect. The relative amount which the water may be cooled below the atmospheric temperature by the evaporation method depends upon a number of factors, such as the temperature of the air, the relative humidity*, and the local conditions. By relative humidity is meant the relative amount of water vapor contained in the air, as compared to the amount that the air can contain when it is completely filled or saturated with water vapor. Under this condition it will be observed that when the air is saturated with water vapor, there could be no evaporation of the water in the earthen jar, under which condition no cooling effect could be produced. On the other hand, when the air contains only a fractional part of the moisture which it may contain, depending upon the temperature, it is evident that it may absorb more water vapor by coming in contact with the small drops clinging to the exterior surface of the walls of the jars. For example, in a locality where the relative humidity is as low as 29 per cent, the theoretical temperature obtained by the evaporation of the water when the air passes over it is 67° F., when the initial temperature of the air is 90° F. This latter temperature of 67° F. is called the wet-bulb temperature, which, as the name indicates, is the temperature obtained on a thermometer, the bulb of which is covered with a piece of soft cloth or wick, that is kept moist with water. Depending upon the local conditions, such as the shape of the earthen jars, their location in reference to the moving currents of air, etc., it will be possible to cool the fluid in the water within a few degrees of the wet bulb temperature which in the above example was 67° F.

Limited amount of cooling effect for purposes other than the cooling of food, may be obtained by evaporation of limited quantities of liquid such as ether and the like.

In the method of cooling by the evaporation of liquids, the refrigerating or cooling effect is produced by the liquid absorbing its latent heat of evaporation, thereby changing its state from the liquid to the vapor state. In order to change the state of the materials from that of the liquid to that of the vapor, it is necessary to supply a certain amount of heat which is dependent upon the individual kind of liquid used, and its relative evaporating temperature.

Cooling by the Melting of Solids.—In this method of producing a cooling or refrigerating effect, the heat absorbed in the process is taken up by a solid form of some substance, changing or melting to the liquid form. In order to change the state of the material in a solid form to the state of the material in the liquid form, a certain amount of heat

* For more complete explanation of properties of air see Chapter XVI.

must be added which is dependent upon the kind of substance used and its relative temperature and pressure. The heat which is necessary to change a material from the solid to the liquid state is sometimes called the latent heat of fusion or melting. One of the solids which is most commonly used for this purpose is ice, which is water in the solid form.

Investigations to determine the exact latent heat of fusion of ice have been carried on by many men at different times. The most authentic researches and tests have been made by the United States Bureau of Standards and a detailed report of the investigations and result

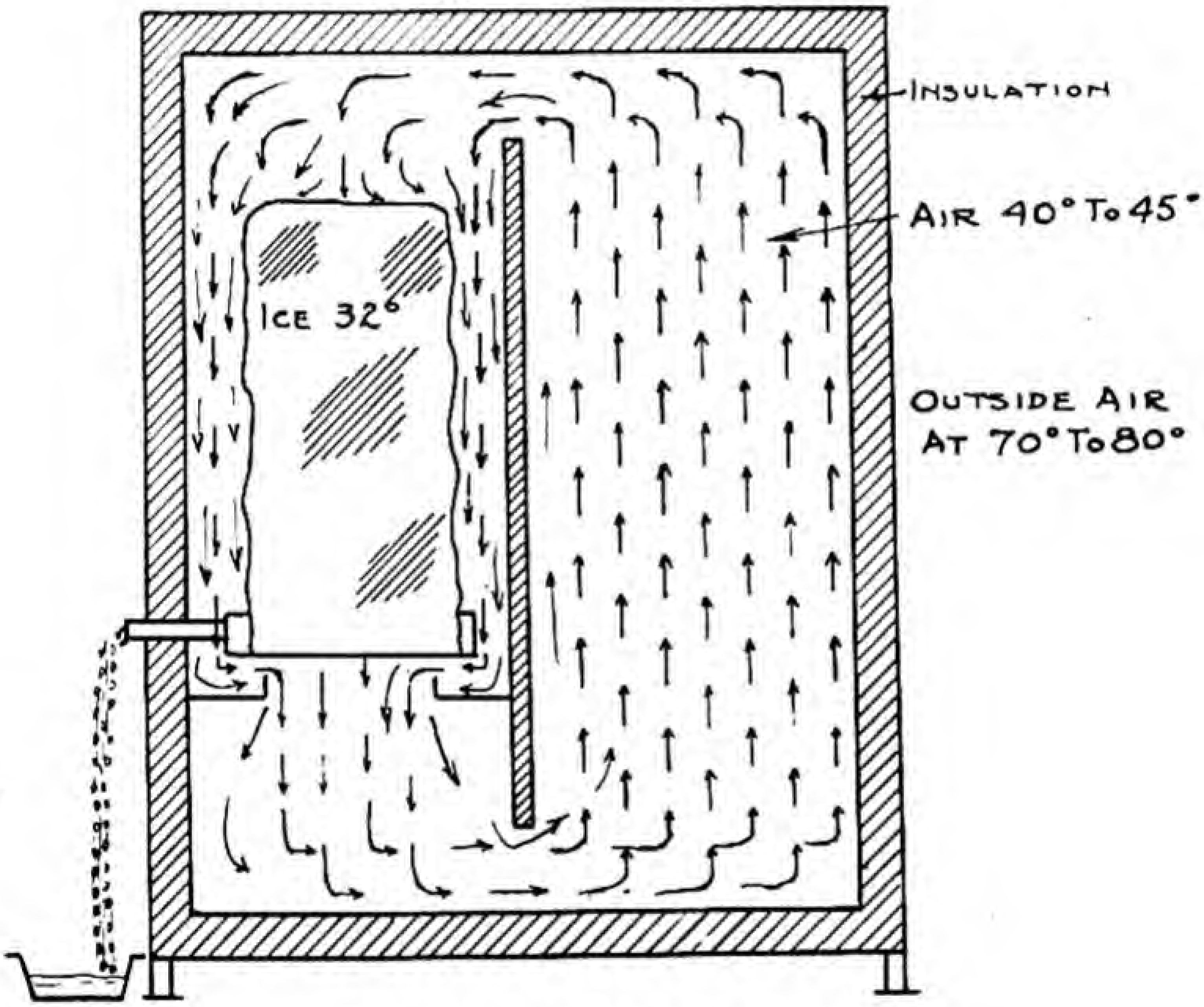


Fig. 2.—Cooling by Melting Ice.

was given in a paper presented by Dr. H. C. Dickinson at the Third International Congress of Refrigeration, held in Chicago in 1913. The American Association of Refrigeration was instrumental in obtaining appropriations from the United States Congress by means of which the Bureau of Standards was able to carry out the necessary research work. The Bureau found the exact latent heat of fusion of ice to be 143.5 Btu. per lb., but for convenience in calculations, a round figure of 144 Btu. per lb. has been generally adopted as a standard. The approximation is very close, however, and is accurate enough for all practical purposes.

In the elementary ice system shown in Fig. 2, the ice absorbs its heat of fusion from the air in the compartment, thereby changing from solid to liquid state, and maintaining at the same time a temperature of 40° to 45° F. in the cooler.

The solid ice which is used for this purpose may be produced by natural or mechanical means, but the refrigerating or cooling effect is practically the same, regardless of the method of producing the ice. The use of solid ice for producing refrigeration or a cooling effect is shown practically by Fig. 2. In this figure is shown an insulated compartment or refrigerator, into which a quantity of ice is placed. The solid form of water or ice has a peculiar and individual characteristic of melting always at a temperature of 32° F. under atmospheric pressure conditions. This property of a low melting temperature, together with the latent heat of fusion, or melting, are taken advantage of in the method of cooling by the melting of the ice as shown by Fig. 2. In this figure, a quantity of ice produced either by natural or mechanical means, is placed in the insulated compartment for the purpose of maintaining a temperature below that of the surrounding air. When the compartment is well insulated, and is supplied with a sufficient quantity of ice, it will be possible to maintain an air temperature inside of the refrigerator from 40° to 50° F. when the air on the outside of the refrigerator is at a temperature of 70° to 80° F. The heat flows by natural tendency from the air on the outside at a temperature of from 70° to 80° F., into the air at a temperature of 40° to 50° F. on the interior of the refrigerator or insulated compartment; the air, coming in contact with the surface of the melting ice, will be cooled. This cooling of the air makes it heavier, in consequence of which, it sinks to the floor of the refrigerator as shown in Fig. 2, allowing additional warmer air to take its place at the surface of the melting ice. On account of the characteristic melting temperature of 32° F. for the ice, the heat will therefore flow from the air at 40° to 50° F., to the melting ice surface which is always at the temperature of 32° F. The resulting water which is formed, due to the melting of the ice, is then removed from the refrigerator by means of a suitable drain pipe. In order to maintain the temperature fairly constant in the insulated compartments, it is evident that the ice supply must be replenished from time to time so that sufficient surface is always exposed to the air in the refrigerator. It is evident that this water which results from the melting of the ice, may be changed back again into solid ice, and this ice reinserted into the compartment for the purpose of absorbing its latent heat of fusion or melting. Of course, due to the fact that water is abundantly distributed, it is not usually considered advisable or practical to refreeze the water coming from the refrigerator.

Cooling by Freezing Mixtures.—In the case of the melting of the ice, 144 heat units (Btu.) must be supplied to each pound of ice in order to change it from solid ice to water at 32° F.

Lower temperatures may be produced in small quantities by certain freezing mixtures. As previously indicated, when ice is used, temperatures as low as 40° to 45° F. may be obtained. When it is necessary to maintain lower temperatures, and when only a small amount of refrigeration is required, the so-called freezing mixtures may be used. One of the most commonly used mixtures consists of salt and ice. The salt is the common salt or sodium chloride and the ice may be cracked or crushed ice, or even snow. The action of the mixture is such that

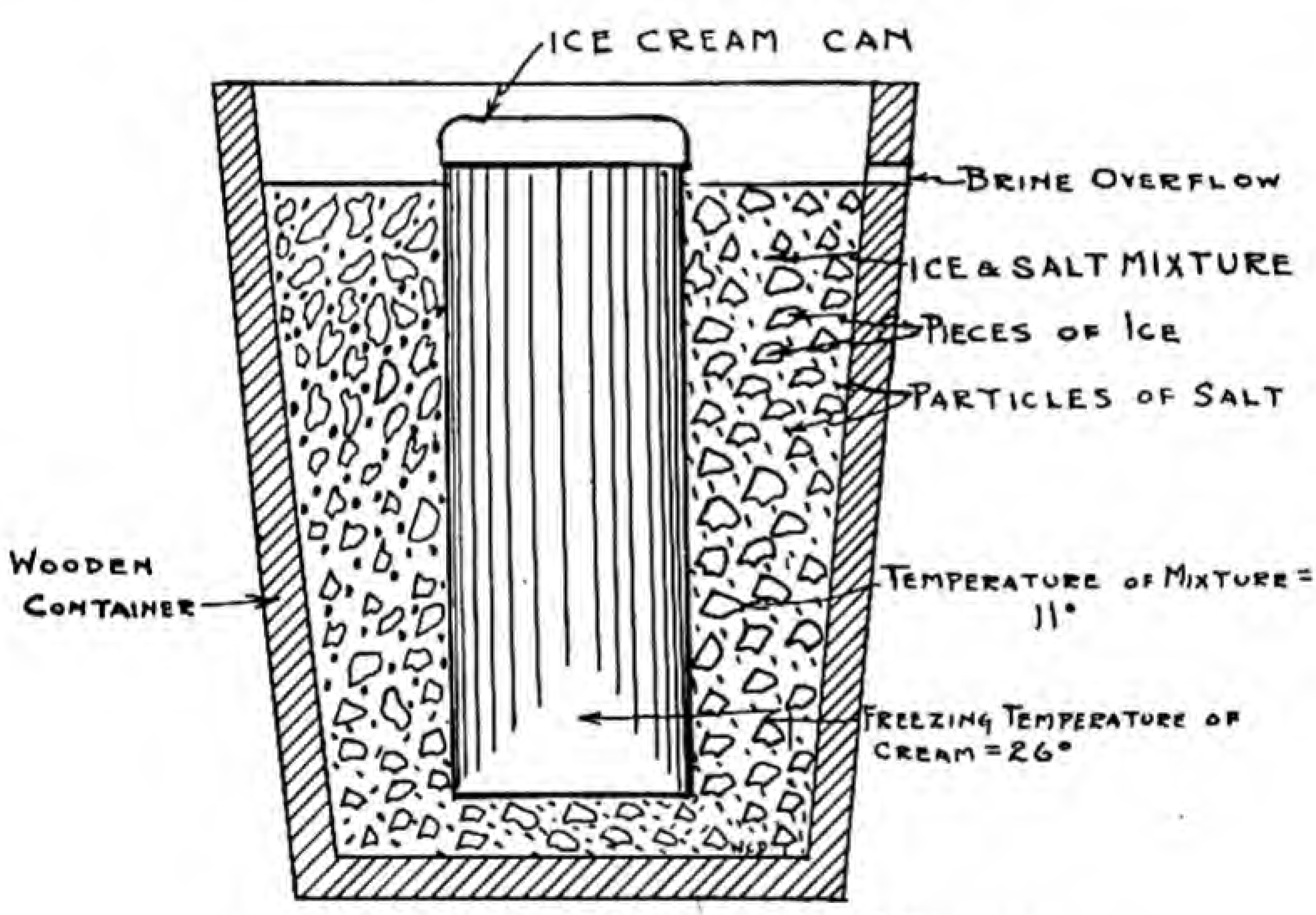


Fig. 3.—Cooling by Freezing Mixture.

the temperature of mixture will be lowered several degrees below 32° F. The actual amount of lowering of the temperature will depend upon several factors, such as the relative proportion of salt and ice in the mixture, the relative size of the ice and salt particles, the rate at which heat is added to the mixture, the relative shape of the container, and the relative amount of brine and solid ice in the mixture. The principal action which causes the reduction of temperature in this method of producing refrigeration may be described briefly as follows:

When two solids such as ice and salt are mixed together to form a liquid, a certain amount of heat is evolved, due to the fact that both undergo a change of state, namely from solid to liquid. The ice will

absorb its latent heat of fusion in melting. In the case of the salt, the latent heat of solution will be absorbed and this varies with the density and temperature of the resulting brine. This latent heat required for dissolving the salt and melting the ice is taken from the mixture itself, thereby cooling the mixture below the temperature of pure solid ice. This method of producing refrigeration is illustrated in Fig. 3, which shows the freezing of ice cream by means of the salt and ice mixture. If the mixture contains 15% salt the resulting temperature of the mixture will be approximately 11° F.

Under this condition, the heat will flow from the ice cream mixture until the temperature has been reduced from the initial temperature of 50° to 60° F. to the freezing point of approximately 26° F. At the temperature of 26° F., the cream will begin to freeze, the heat flowing from the cream at an approximate temperature of 26° F. into the freezing mixture at 11° F. Increasing the proportion of salt used in the mixture, lowers the freezing point correspondingly. Other substances may be used in a similar manner for producing freezing mixtures. For example, if two parts of snow are mixed with three parts crystals of calcium chloride, the temperature of the mixture will fall from 32° F. to approximately −50° F. Mixtures of solid carbon dioxide and acetone are used for temperatures below −70° F.

Cooling by Sublimation.—Sublimation is the changing of a solid to vapor state without passing through the intermediate state of liquid. Due to the change of state latent heat will be absorbed during the sublimation process, and hence refrigeration may be produced.

Solid carbon dioxide is an example of this process and is used for producing refrigeration for the shipping and holding of frozen products.

The carbon dioxide is used in the cake or block form. It has a temperature of −109° F. or lower at normal atmospheric pressure. Instead of melting to liquid as water ice does, the solid carbon dioxide sublimes, that is, passes directly from solid to vapor state. This is one of the advantages of this substance when used as a refrigerant as it does not wet packages or materials refrigerated with it.

Evaporation of Liquids.—Practically all mechanical refrigeration systems utilize the evaporation of some liquid refrigerant at a low temperature. During evaporation the liquid is converted into a vapor. The various direct systems in use differ only in the method of restoring the vapor to its liquid state. The names of these systems suggest the method used in the restoration process rather than the method of producing refrigeration.

An elementary evaporating system using a volatile fluid is indicated in Fig. 4. A container for holding the refrigerant is put into the insu-

lated compartment as shown. The refrigerant may be any of the commonly used media such as ammonia. If the ammonia in the container is subjected to the pressure of the atmosphere as indicated in Fig. 4, the temperature of the ammonia vapor as well as the liquid is —28° F. under this condition, and then the compartment may be maintained at 0° F. while the temperature on the exterior may be 80° F. The heat from the air in the interior of the compartment flows by natural tendency into the ammonia, which causes it to boil or evaporate out of the container. A constant temperature of —28° F. exists during the process of evaporating.

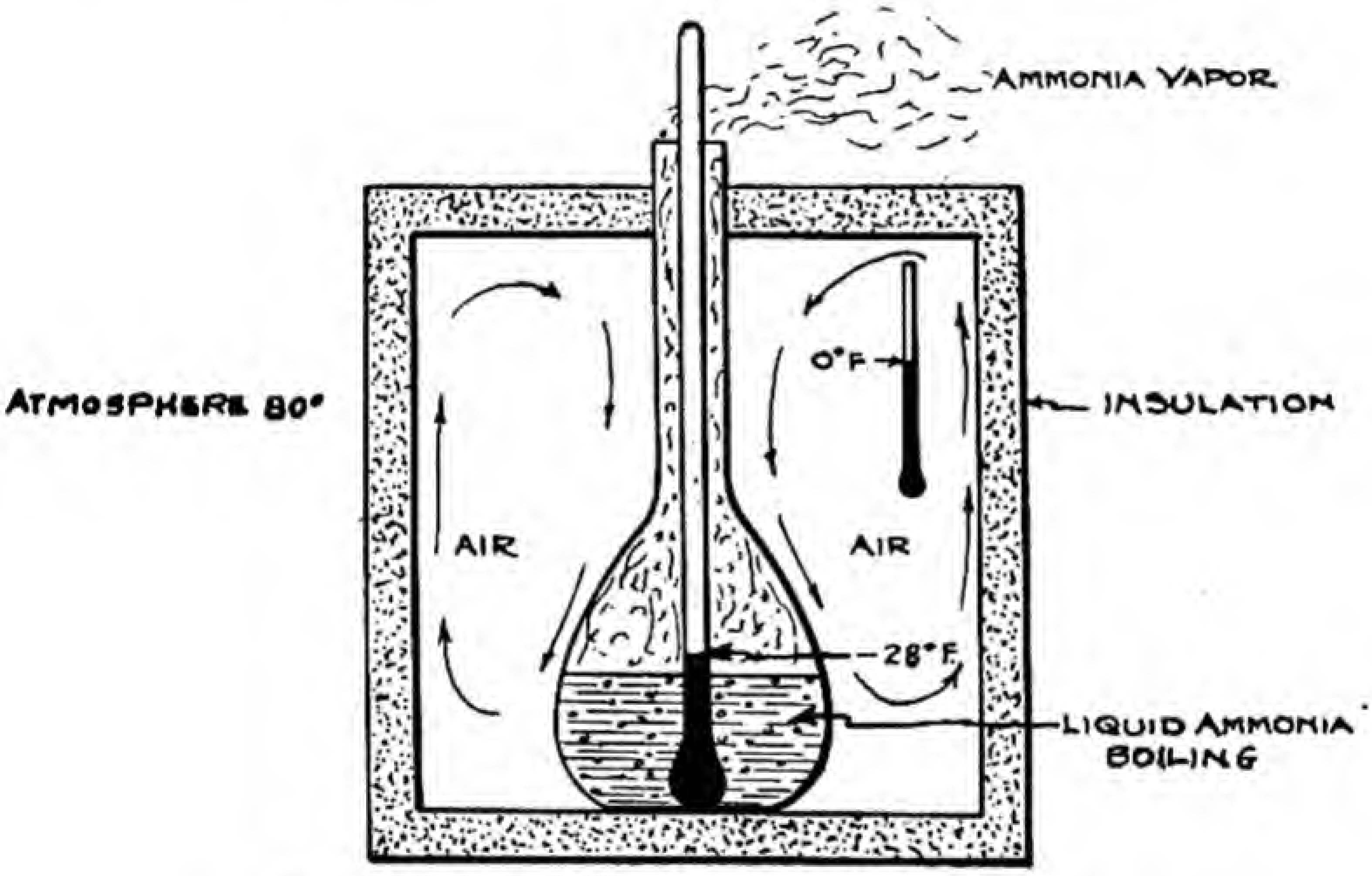


Fig. 4.—Elementary Evaporating System.

As previously indicated, when heat is added to a substance without change of temperature, a change of state is produced, such as liquid to vapor, as in this case. The heat required to produce the change from liquid to vapor state is called the latent heat of evaporation.

In the case of ammonia liquid at —28° F., corresponding to atmospheric pressure, 589.3 Btu. are required to completely vaporize each pound. This 589.3 Btu. is known as the latent heat of vaporization.

Thus in the elementary direct expansion system the heat which passes through the insulation together with the heat removed from the materials stored in the cooler are transmitted by the air to the boiling ammonia in the vessel. The ammonia absorbs its latent heat at the low temperature and thereby produces the desired refrigerating effect. In the system shown by Fig. 4 the resultant vapor from the boiling refrigerant is allowed to escape to the atmosphere.

If the ammonia liquid had been introduced into the vessel at a temperature higher than $-28°$ F., for example, $80°$ F., then part of its latent heat of vaporization (119.2 Btu. per pound) would have been used to cool itself down to $-28°$ F. and only the balance 470.1 Btu. per pound would have been available for producing refrigeration.

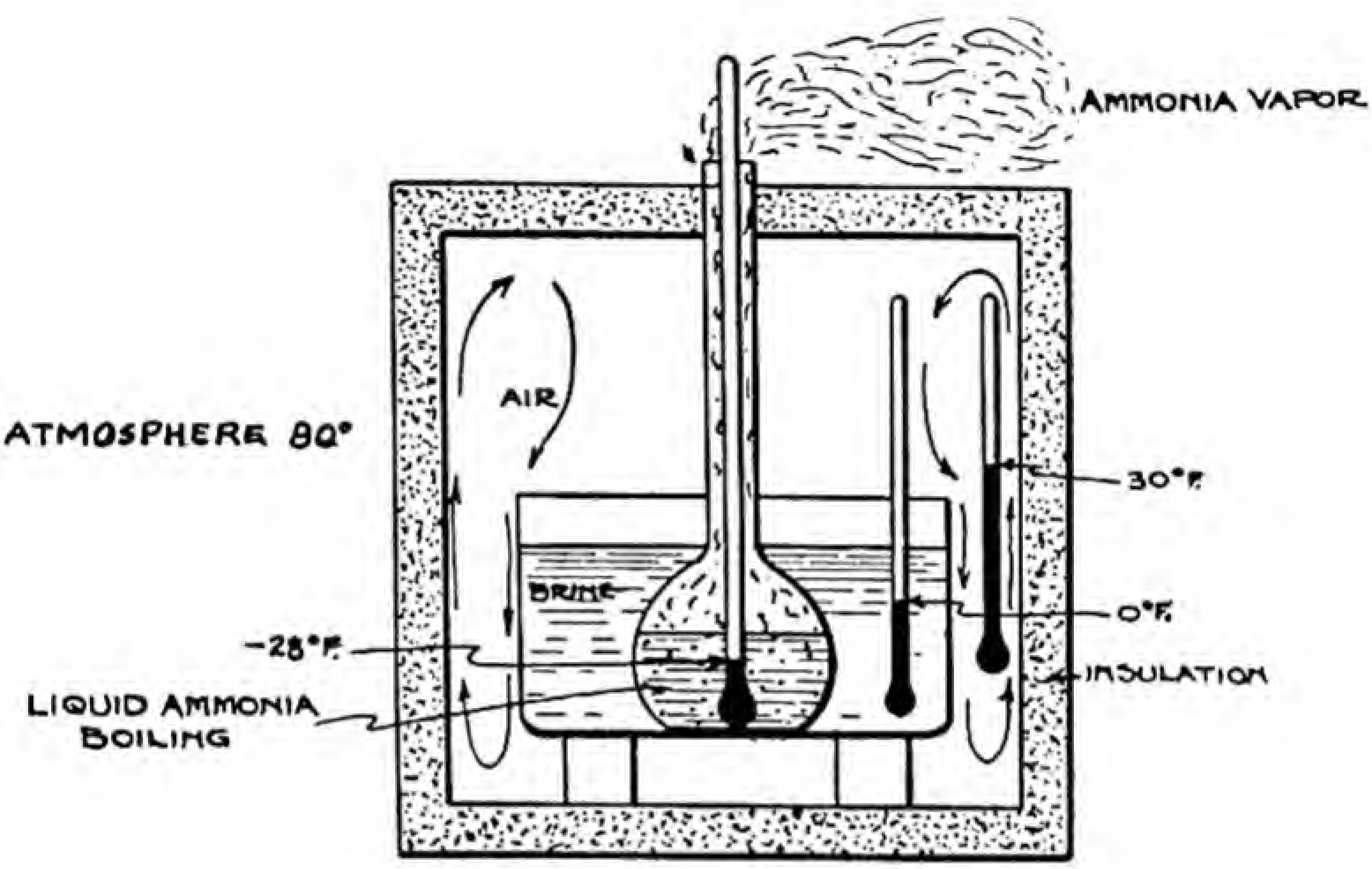


Fig. 5.—Elementary Brine System.

Under certain conditions it is desirable to make use of the brine system of refrigeration. An elementary brine system is illustrated in Fig. 5. The general principle underlying the operation of this system is the same as that of the elementary direct expansion system.

To maintain the insulated compartment at $30°$ F. a vessel containing liquid ammonia is inserted into a quantity of brine which cools and maintains the brine at $0°$ F. The heat from the atmosphere at $80°$ F. on the outside is transmitted through the insulated wall to the air in the box at $30°$ F. This heat in turn is transferred by the air and brine to the boiling ammonia at $-28°$ F. causing it to evaporate. The temperature differences between the atmosphere and air in the compartment, between the air and the brine, and between the brine and the boiling ammonia causes the necessary amount of heat transference.

An elementary ice freezing system is shown in Fig. 6. A vessel containing liquid ammonia or other refrigerant is inserted into a quantity of brine. The ammonia, being exposed to the pressure of the atmosphere has a temperature of $-28°$ F. and may maintain the brine at $12°$ F. On account of this reduced temperature, the water surround-

ing the brine container freezes on the surface of same, as shown by Fig. 6. The ammonia vapor escapes to the atmosphere after having absorbed its latent heat of evaporation from the brine, which in turn absorbs the latent heat of fusion from the water. This causes the water to solidify into ice. The brine also absorbs the heat required to cool the water to the freezing point as well as the heat transmitted by the insulation.

Compression Refrigerating System.—This system utilizes a mechanically driven compressor to withdraw the low temperature vapor from the evaporator and compress it to a pressure sufficiently high when it may be condensed by water, air, or other economical mediums

The complete cycle may be divided into four principal phases:

(1) Evaporation of liquid at low temperatures.

(2) Compression of vapor.

(3) Condensation of vapor at high temperatures.

(4) Throttling of liquid.

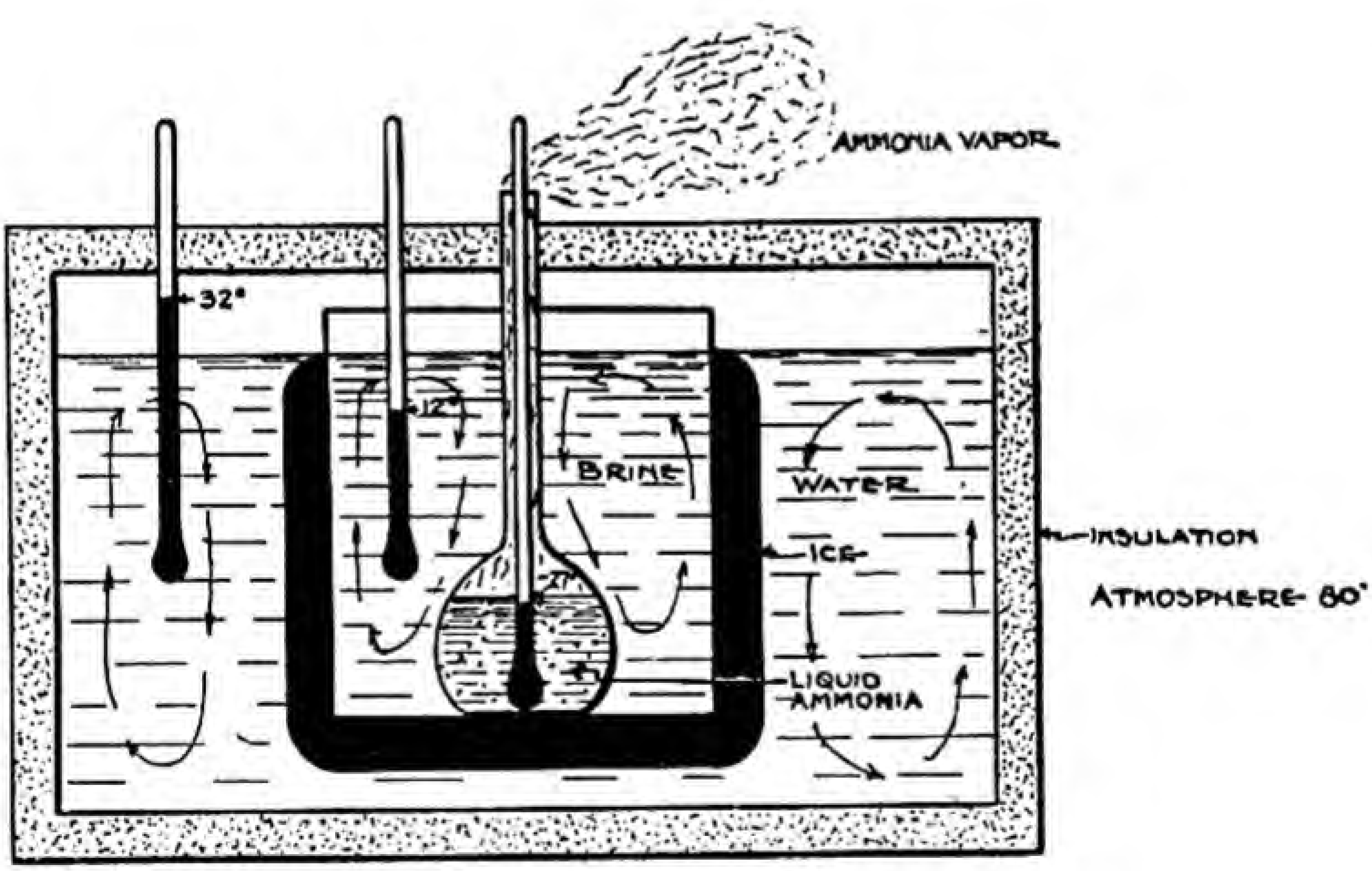


Fig. 6.—Elementary Ice Freezing System.

The essential parts of the compression refrigerating system are shown by Fig. 7. The compression refrigerating system shown in Fig. 7 uses ammonia as a refrigerating medium. If the pressure is maintained in the evaporator at 19.6 lbs., the characteristic boiling point of ammonia at this pressure is 5° F.; under which condition, the

heat flows from the air in the refrigerator at the average temperature of 30° F. into the ammonia, causing it to be evaporated. A certain amount of heat must be added to each pound of liquid ammonia to change its state from the liquid to the vapor form. This heat comes from that which is transmitted through insulation, or otherwise, into the room, and is finally transmitted to the evaporator surface by means of the circulation of the air in the room. The ammonia vapor is withdrawn from the evaporator as fast as it is produced by a mechanically operated piston compressor. The type shown in Fig. 7 is the single-acting type in which the down stroke is the suction stroke and the up stroke is the compression or discharge stroke.

Another characteristic property of ammonia, as well as other liquefiable fluids, is that its condensing pressure rises in some proportion to the increasing of the temperature. Conversely, it may be stated that the evaporating temperature will be lowered in some proportion to the lowering of the suction pressure. In the case of the evaporator, it is evident that the temperature of the boiling ammonia must be maintained a few degrees below the average temperature in the refrigerator, so that the heat will flow from the materials in the refrigerator into the ammonia, causing it to be evaporated, thereby producing the desired refrigerating effect. On the other hand, it will be observed that the heat absorbed in the refrigerator, together with heat absorbed during the process, must be discharged at some higher temperature. The material which is most commonly used for heat taken out of the refrigerator, and the heat added during the process, is water. Air is used sometimes. Due to the fact that the temperatures of water supplies are in nearly all cases above the temperatures desired in the refrigerator, it is evident that the pressure of the ammonia must be changed or increased so that its condensing temperature is a few degrees above the average temperature of the water supply.

This fact establishes the function of the compressor, which may be briefly stated as follows:

The compressor raises the pressure of the ammonia from the pressure corresponding to the evaporating temperature in the refrigerator, to the pressure corresponding to the condensing temperature in the condenser. If it is supposed that a water supply is available at a temperature of 70° F., and that the water in passing through the condenser rises to 80° F., it is possible to maintain a condensing temperature of 86° F. for the ammonia in the condenser. The characteristic pressure corresponding to this temperature for ammonia is 154.5 lbs. gauge. The function of the compressor, therefore, is to withdraw the ammonia vapor of the evaporator as fast as it is formed at the pressure of 19.6 lbs. gauge and to compress this resulting vapor to a pressure of 154.5 lbs. per sq. in. gauge. It is evident that the pressure and temperature

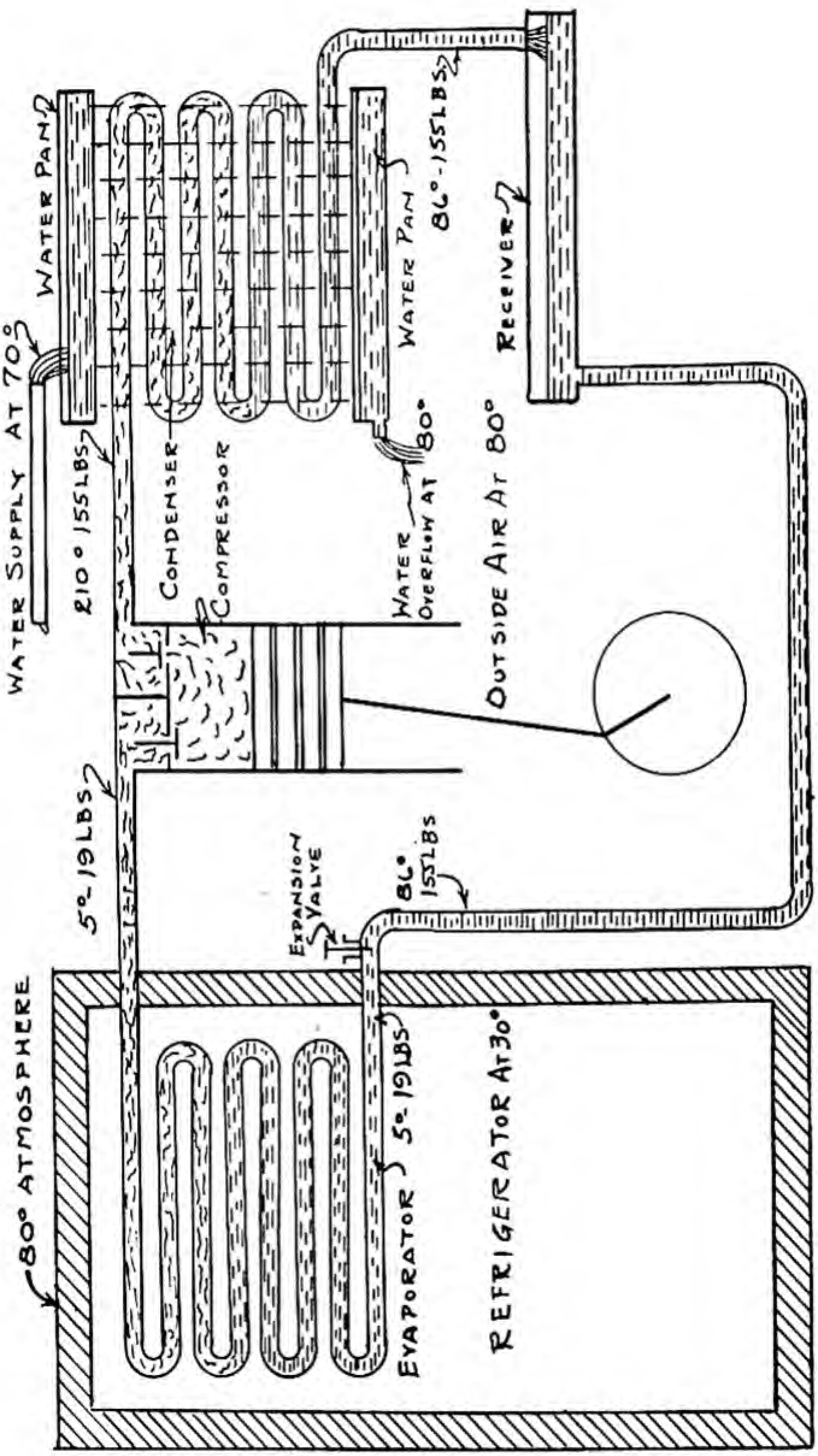


Fig. 7.—Compression Refrigerating System.

in the evaporator will depend upon the temperature desired in the refrigerator, and that the pressure and temperature in the condenser will depend upon the temperature of the available water supply. When the evaporating pressure is 19.6 lbs. per sq. in. gauge, and the ammonia comes to the compressor in the dry or saturated state, that is without suspended particles of liquid, the temperature after compression will be approximately 210° F. The function of the condenser, therefore, is to cool the hot ammonia gas from a temperature of 210° F. down to the condensing temperature of 86° F. In the condensing zone of 86° F., the heat goes from the ammonia at this temperature into the water at a lower temperature, until the latent heat of condensation has been transferred into water, thus changing the vapor ammonia to the liquid state. The liquid ammonia at the temperature of 86° F. is led from the condenser to a suitable liquid receiver which acts as a storage tank. This storage tank or receiver is connected by a suitable pipe line connection to the so-called expansion or throttle valve. The pressure and temperature of the liquid ammonia just before the ammonia valve is 154.5 lbs. per sq. in. gauge and 86° F. respectively. The so-called expansion or throttle valve serves the purpose of reducing the pressure from the high pressure existing in the condenser to the low pressure existing in the evaporator. The liquid ammonia has been therefore returned to the evaporator in the liquid state to be re-evaporated and used over and over again. This completes the four principal phases of the compression refrigerating cycle. The system is represented diagrammatically by Fig. 7 and the compressor may be run by any suitable prime mover. The compression refrigerating system, as illustrated in Fig. 7, will maintain a temperature of 30° F. in the insulated compartment when the atmospheric temperature is 80° F. It can be used in a similar manner, to maintain other temperatures ranging from —25° F. to 65° F. in the refrigerated compartment. It is made in sizes ranging from a small fraction of a ton of ice melting effect per day to as large as 1000 tons of refrigeration per day.

Absorption Refrigerating Machine.—The absorption refrigerating machine has been extensively used for the production of refrigeration where especially low temperatures are required, or where there has been a considerable quantity of waste heat in the form of steam available. The absorption refrigerating system is shown diagrammatically by Fig. 8, which illustrates the principal parts of this type of refrigerating machine. In this system, the condenser, expansion valve, receiver, and evaporator, are of the same type that is used in the compression refrigerating machine. The systems differ primarily in reference to the method of extracting the vapor from the evaporating coils and discharge of same in condenser.

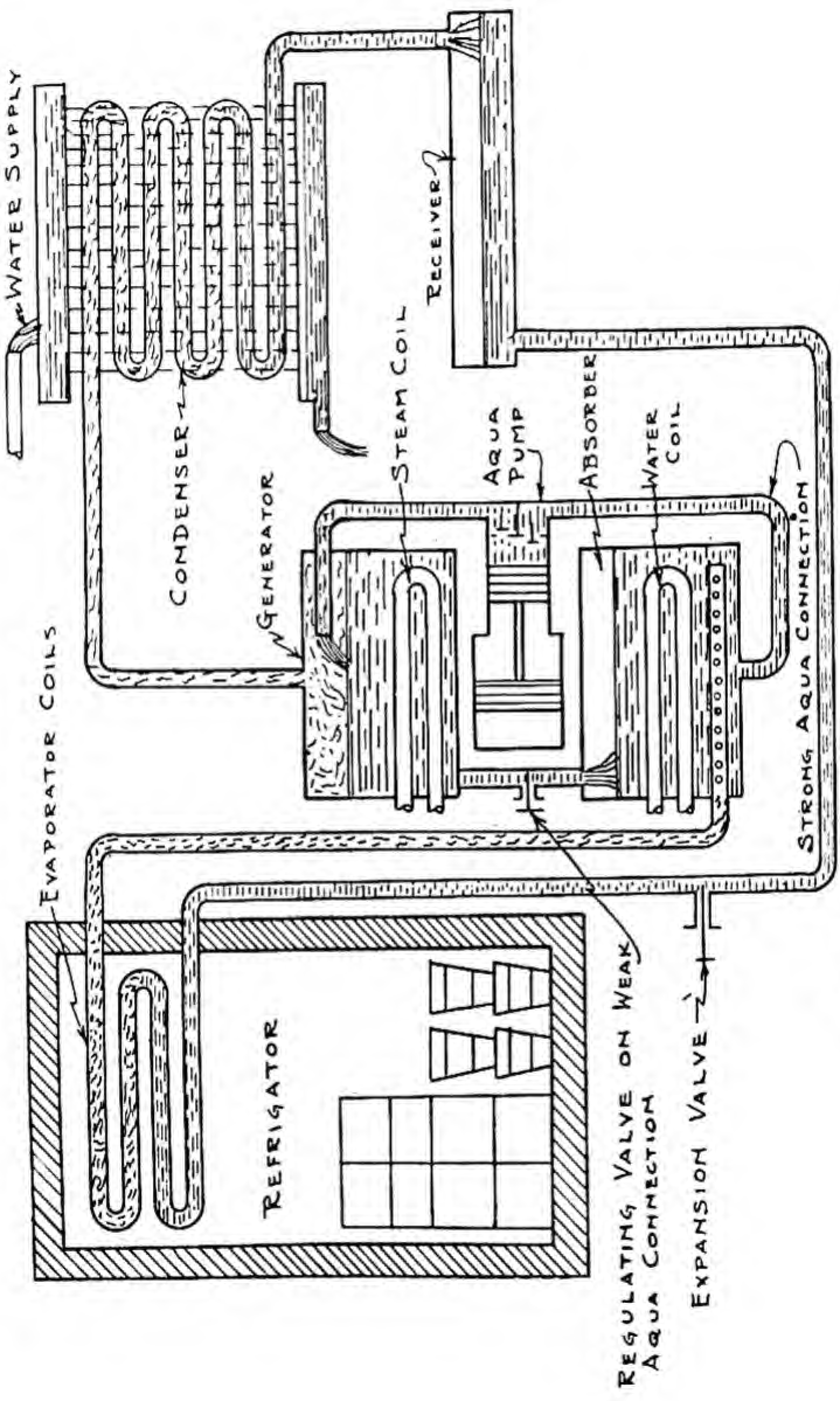


Fig. 8.—Absorption Refrigerating System.

The vapor from the evaporating coils is removed from same by dissolving it in a weak solution of ammonia in water. Water has a great affinity for ammonia, dissolving within itself, many times its own volume of ammonia. This part of the system is called the absorber, shown in Fig. 8.

The absorber consists primarily of four elements: The shell for retaining the solution of ammonia in water; a device for introducing the vapor from the evaporator into the aqua ammonia solution; a water coil for removing heat from the absorber: and strong and weak aqua connections. The ammonia from the evaporator is led to the absorber in a vapor form. It is first condensed and then dissolved in the weak aqua ammonia solution in the absorber.

This change of state liberates latent heats, which are removed by water flowing through the water cooling coil, thus maintaining the absorber at a uniform temperature. The temperature of the solution in the absorber will always be a few degrees above the average temperature of the water in the coil.

The percentage of ammonia which the solution will absorb depends upon the relative evaporating pressure and the temperature which is maintained in the absorber. When the solution has absorbed all the ammonia it is able to hold at the pressure and temperature, it is then led through a pump which discharges the strong aqua ammonia into the generator, or distilling apparatus. The pressure in the absorber corresponds very closely to the pressure in the evaporator, while the pressure in the generator corresponds very closely to the pressure in the condenser.

The function of the pump is therefore simply to remove the strong aqua from the absorber and discharge it into the generator. By introducing steam into the steam heating coils in the generator, it is possible to bring the solution to the boiling point, and to boil it, thereby distilling the ammonia vapor (and some water vapor from the solution). The temperature of the boiling solution in the generator will be determined by the relative condenser pressure and the percentage of the ammonia in the weak aqua ammonia as it leaves the generator to go back to the absorber.

The temperature of the steam in the steam heating coil must always be a few degrees higher than the boiling solution in the generator, so that the heat will flow from the steam into the solution, causing it to boil and thus driving off the ammonia vapor. The ammonia vapor then passes on to the condenser, where it is condensed, and then returned to the expansion valve in liquid form.

The Vacuum Process.—In the vacuum process, water is made to boil at a low temperature and pressure and thus produce refrigeration

or ice. For example, water at an absolute pressure of 0.0886 lbs. will boil at 32° F. The volumes of water vapor to be handled are large, since a pound of water vapor or steam will occupy 3,294 cu. ft. of space at an absolute pressure of 0.0886 lbs. per sq. in.

These large volumes of water vapor may be handled by a vacuum pump, but owing to mechanical difficulties, due to large sizes and high vacuum, the water vapor is sometimes absorbed in a good absorbent such as sulphuric acid. The acid becomes saturated with the water, may be reconcentrated by reboiling, and then used over and over again. Such machines are not used very extensively.

The Adsorption System.—The adsorption system is illustrated by the silica gel refrigeration system. It usually operates on the intermittent cycle, similar in effect to the intermittent absorption machine.

Silica gel is a hard, glassy material of extremely porous character. The presence of the minute voids or pores gives silica gel the ability to adsorb relatively large amounts of vapors or gases. After adsorption of the vapor or gas, the silica gel may be activated by heating, thus driving out the vapor and rendering it capable of re-adsorbing vapor again. This action is entirely physical and may be repeated indefinitely.

Essentially the apparatus consists of four principal parts, namely: the adsorber, containing silica gel; the evaporator; the condenser; and the pressure-reducing element. Briefly the system is similar to the compression cycle. The conventional compressor is replaced by the adsorber. The adsorption of the refrigerant vapor by the silica gel corresponds to the suction stroke; the activation corresponds to the discharge stroke. No power is required for the silica gel or adsorption system. The necessary heat is applied directly to the silica gel.

Indirect Air Refrigerating System.—The indirect air refrigerating system is shown diagrammatically by Fig. 9. This system consists primarily of a bunker room, containing evaporator coils for cooling the air, air circulating fan and connections, and air distributing devices in the refrigerated room and the bunker room.

The evaporating coils may be connected to any type of refrigerating machine which produces an evaporating temperature that is a few degrees below the average temperature of the air circulated. The air is thus cooled a few degrees in passing through the bunker room. It is then led to the refrigerator or cooler, where it absorbs heat, rising in temperature in proportion to the drop in temperature in the bunker room. Thus, the air, by absorbing sensible heat, will maintain the refrigerated room and its contents at a low temperature. Of course, the air must be circulated by means of a power driven fan or blower, but

when the piping system is correctly laid out, the power required for circulating the air will be reduced to a minimum.

One of the principal advantages of the indirect air refrigerating system is that it is possible to maintain a fairly uniform temperature in all parts of the refrigerated room.

Indirect Brine Refrigerating System.—In some refrigerating installations, it is not advisable or desirable to install the refrigerant evaporating coils in the refrigerators or coolers directly, or where the

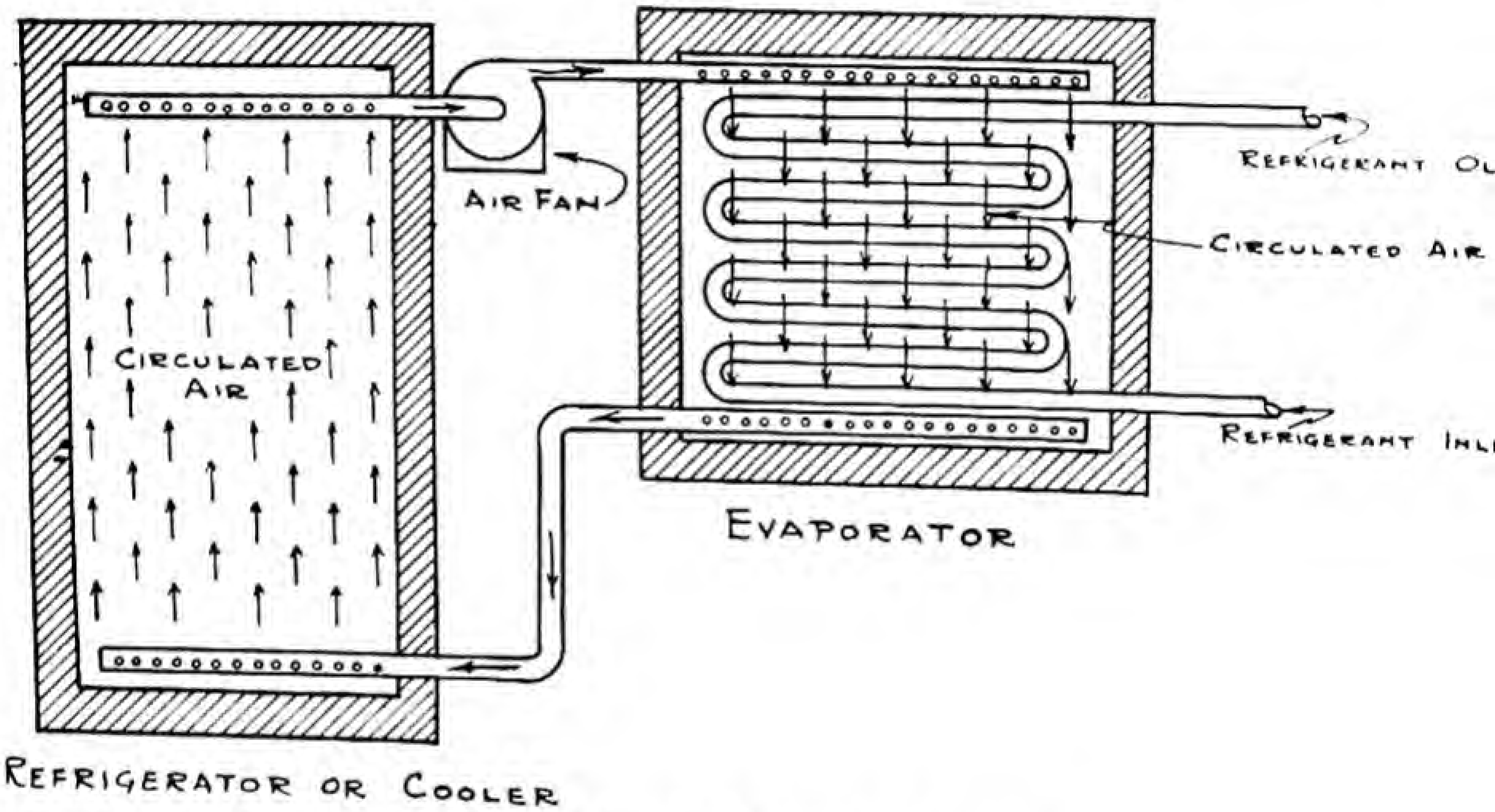


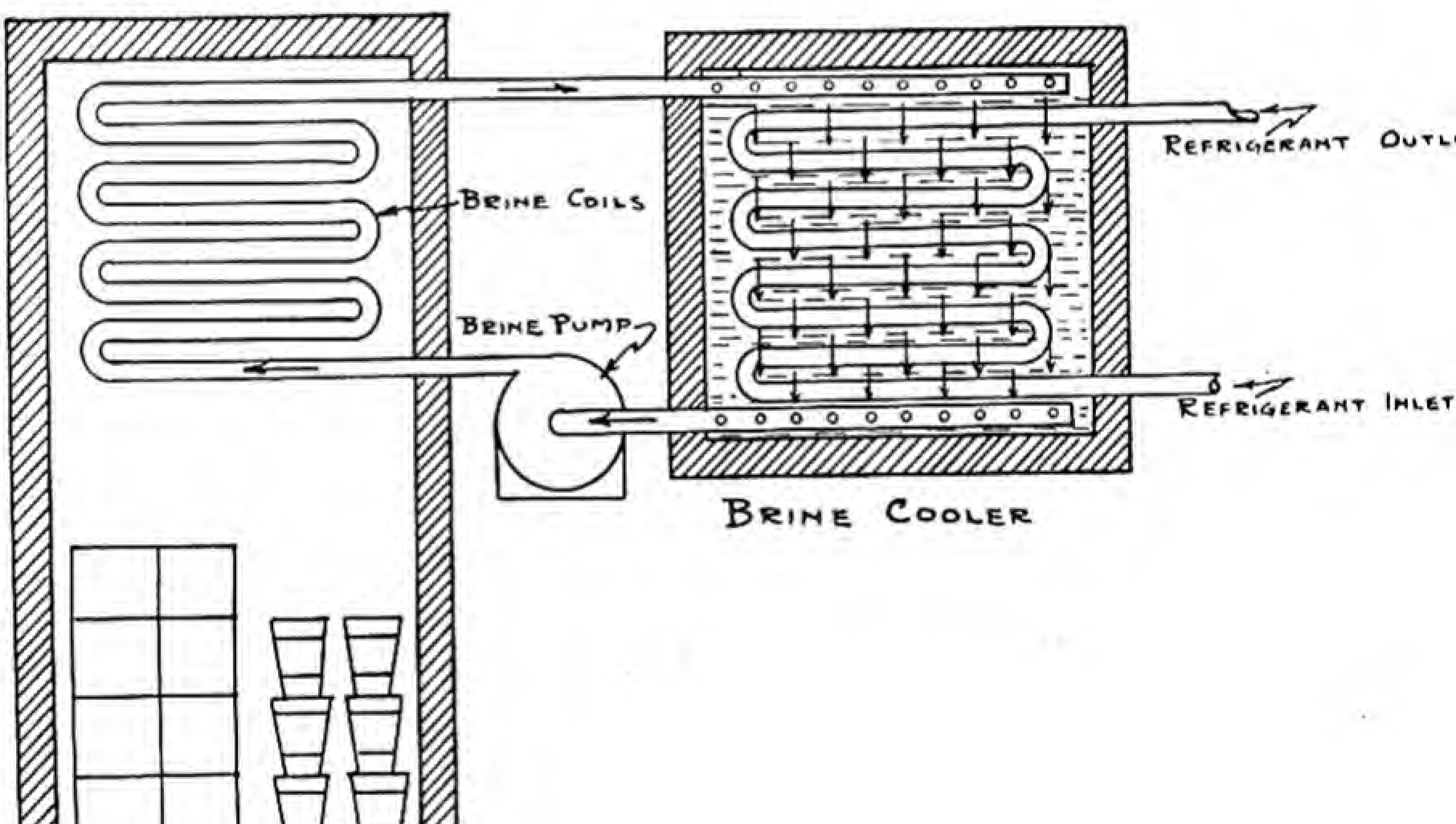
Fig. 9.—Forced Air Refrigerating System.

application of the indirect air refrigerating system is not feasible. The indirect brine refrigerating system is shown diagrammatically by Fig. 10. It consists of a brine cooler for cooling the brine, a pump for circulating the brine, brine coils in the refrigerated rooms, together with suitable brine connections. The brine cooler coils may be connected to any suitable refrigerating machine which will maintain a sufficiently low evaporating temperature in the coils.

The temperature of the evaporating refrigerant must always be a few degrees below the average temperature of the brine so that the heat will flow from the brine into the refrigerant. The brine, after being cooled, passes into a pump, and thence into the refrigerating coil in the refrigerator.

In this refrigerating system, the brine must always be a few degrees below the average temperature of the air in the refrigerated room, so

that the heat will flow from the refrigerator into the brine, thus maintaining the refrigerated room at a certain low temperature. Thus, the heat must be transferred, first from the air to the brine in the refrigerated room, and then from the brine to the evaporating refrigerant in the brine cooler. In this system, it is therefore necessary to maintain two temperature differences between the evaporating refrigerant and the temperature in the refrigerator. Consequently, it is necessary to maintain a slightly lower refrigerant evaporating temperature, with this system, than when the evaporating coils are introduced into the refrigerated room directly; maintaining a lower refrigerating evaporating temperature means that the power requirements would be increased and the compressor size will also be increased when the compression refrigerating system is used.



Refrigerator or Cooler

Fig. 10.—Brine Refrigerating System.

The advantages of the system are that it is flexible in operation, and that it has certain safety features, due to the elimination of the refrigerant from the refrigerator room directly.

Brine Spray System.—The brine spray system, as the name indicates, is a system in which the brine is sprayed directly in the air to be cooled. The brine spray system is shown diagrammatically by Fig. 11.

The system consists of a brine spray header, to which is attached several brine spray nozzles, a brine proof bunker, a brine pump, brine cooler, and suitable brine connections. Passing through the spray nozzle, it is in the form of fine drops traveling through the air, and before they fall to the bunker floor, they are heated a few degrees, thus absorbing heat from the air in the room. The air thus becoming cooler, becomes heavier and falls down through the cold air duct, to the room below. The air heats a few degrees in passing through the room or cooler, and then passes up through the warm air duct to the warm air loft, to be recooled by again passing through the brine sprays. The brine is cooled in a manner similar to the indirect brine system previously described. The usual connections are as shown in Fig. 11, that is, the brine is discharged in the brine spray loft, and led to a brine pump, which discharges the brine through the brine cooler, and then through the brine connections to the brine spray headers. In this system, the temperature of the evaporating refrigerant in the brine cooler must be maintained a few degrees below the average temperature of the brine, and the brine temperature must be maintained a few degrees below the average temperature of the air in the refrigerator.

The brine cooler cools the brine the same number of degrees as it has been previously heated in passing through the air in the cooler. The pressure in the brine spray header must be maintained at 5 or 10 lbs. per sq. in. gauge, to produce the proper spraying of the brine. This system is used principally in meat coolers in packing houses.

Indirect Water Refrigerating System.—In many installations such as air cooling and conditioning, milk and cream cooling, and other similar applications of refrigeration of a fairly high temperature, use is made of the indirect water refrigerating system. This system operates in the same manner as the indirect brine refrigerating system, the water being cooled and circulated in practically the same manner.

In applications requiring temperatures above 32° F. the water may be sprayed directly into the air to produce the necessary cooling effect, just as brine is sprayed for lower temperature work.

The water may be cooled by any of the conventional type of coolers, such as shell-and-tube coolers, coils and tanks, or overflow Baudelot type coils. In the case of the use of shell-and-tube water coolers, provision must be made for protecting the coolers against freezing, by preventing the refrigerant temperature from becoming too low.

Compressed Air Machines.—In the compressed air machine, air is compressed by a driving engine, it is then cooled, and then expanded in an expansion cylinder which does external work. This causes the air to leave the cylinder at low temperature. This cold air is then introduced into space to be cooled, and is heated to the room temperature

before leaving same, thus performing useful refrigerating work. This is the open cycle of operation. For example, if the air is taken into the compressor at 59° F. and compressed to 52.5 lbs. gauge and then is cooled to 64° F., and is then expanded back to atmospheric pressure,

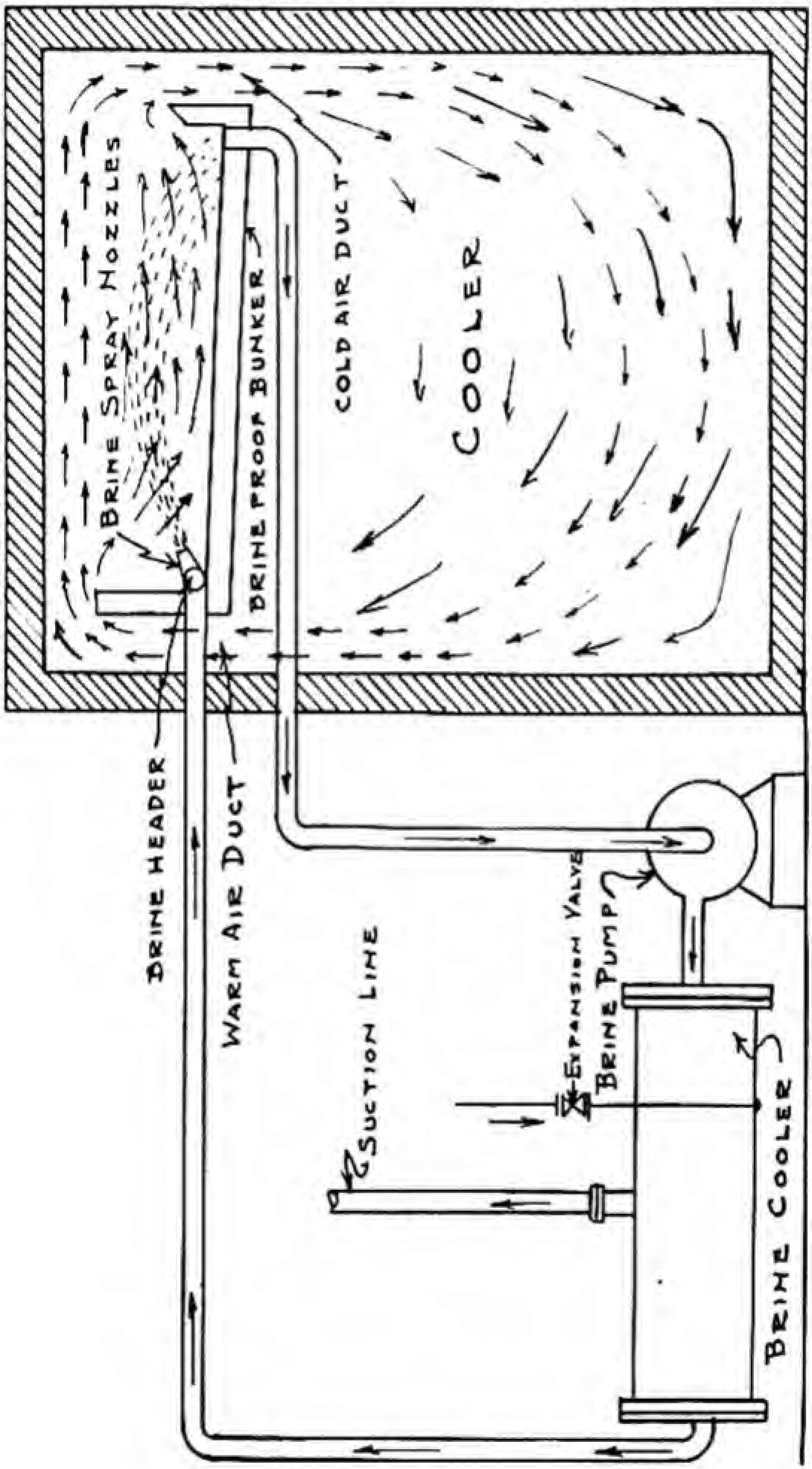


Fig. 11.—Brine Spray System.

its final temperature will be about $-121°$ F. If the room is to be maintained at 32° F., the air will be introduced into the room at a temperature considerably below the temperature of 32° F. and will thus perform refrigerating work by absorbing its sensible heat.

In other arrangements of the system the air is used over and over

again. In this case, the cold air is circulated through pipes located in the rooms to be refrigerated. Generally in the closed system the air is compressed to about 150 lbs. gauge pressure and is circulated through the room piping at about 50 lbs. gauge pressure.

Due to the fact that only sensible heat is absorbed by the air, large volume must be circulated per unit of refrigeration, as compared to a machine using a liquefiable fluid. Due also to the size and friction of the machine, the horsepower per ton of useful refrigeration is larger than where a liquefiable fluid is used.

Due to these and other factors, the use of compressed air machines is limited to more or less special installations.

QUESTIONS ON CHAPTER I.

1. What are the principal methods of producing refrigeration?
2. In a refrigerating system employing the evaporation of a liquid, explain how the desired refrigerating effect is obtained.
3. An ice refrigerator absorbs 3,000 Btu. per day. What is the cost of ice for the refrigerator for 180 days, if the ice costs 60 cents per 100 lbs.?
4. Why is the temperature of the mixture lowered, when ice and salt are mixed together?
5. Describe the action of the refrigerating fluid in a compression refrigerating system.
6. In a compression refrigerating system, why is it necessary to maintain two different pressures on the refrigerant in the system?
7. What are the principal differences between the compression and absorption refrigerating systems?
8. Describe the action of the indirect air refrigerating system.
9. Describe the brine refrigerating system, giving its advantages and disadvantages.
10. Explain the operation of the adsorption system.

CHAPTER II.

FUNDAMENTAL UNITS AND PHYSICAL LAWS.

General Considerations.—Engineering may be defined as the systematic application of science to the economic production of commodities. It is the transformation and utilization of the great stores of energy in nature for the purpose which will serve mankind best. The form of energy that is most useful is that form of energy which is termed work or mechanical energy. The forms of energy as they occur in nature are seldom those which are immediately available for the use of mankind. The needs of modern organized society are such that mankind is obliged to avail himself of the chemical energy of combustible substances. In order to utilize the vast stores of energy in nature it is necessary to transform the combustible substances into heat. This heat, in turn, is transformed into a useful form of energy such as work, electricity or light. Then, by means of these useful forms of energy, various commodities, such as ice and refrigeration, are produced for the service of mankind.

Thus it is the province of the refrigeration engineer to apply in a successful manner the basic principles of physical sciences such as mechanics, thermodynamics, hydraulics, physics, chemistry and heat engineering to the economic production of refrigeration.

As reference will be made from time to time to fundamental physical laws, it is necessary to define the various units of measurement that are used to express the magnitude of the various quantities, giving special attention to those units which are employed in refrigeration practice.

Length.—The unit of length is the foot, and is defined as being 0.30480 meter. The meter is the length of a certain bar of metal, which has been accepted as the standard unit of length by international agreement. In certain engineering work, of course, divisions and multiples of the foot are used as units of measurement.

Mass.—The unit of mass, or quantity of matter, is the pound, and is defined as being 0.45359 kilogram. The kilogram is the mass con-

tained in a certain bar of metal, which has been accepted as the standard of mass by international agreement.

Time.—The unit of time is the mean solar second, simply called the second. This second is defined as being the 1/86400 part of mean solar day.

Force and Weight.—Weight is the attraction of the earth upon matter. The unit of force is weight of one pound of mass, at sea level. For convenience and for brevity the unit of force is known as a pound generally.

Area.—The unit of area is the square foot. However, in many practical engineering problems the square inch is used, which is the 1/144 part of a square foot.

Volume.—The unit of volume is the cubic foot. But in some calculations the cubic inch is used to an advantage. The cubic inch is the 1/1728 part of a cubic foot.

Example 1.—The displacement of the plunger of a brine pump having a 6-in. diameter and a 12-in. stroke is required. Since the area of a circle is equal to the diameter multiplied by the diameter and this product in turn multiplied by the constant 0.7854, the area of piston is $0.7854 \times 6 \times 6 = 28.27$ sq. in.; the volume of cylinder is $28.27 \times 12 = 339.24$ cu. in., not considering the effect of the piston rod. Now, if the pump makes 60 working strokes per min., the piston displacement will be $339.24 \times 60 = 20354$ cu. in. One U. S. Gallon contains 231 cu. in., so that the theoretical displacement is $20354 \div 231 = 88.1$ g.p.m. The actual delivery of the pump will be somewhat less owing to the effect of the leakage past the piston and valves, and to imperfect filling of the pump cylinder during the suction stroke.

Example 2.—The number of cubic inches of piston displacement of a 15-in. dia. $\times$ 30-in. stroke horizontal double-acting compressor cylinder having a piston rod 3 in. in dia. and operated at 70 r.p.m is to be determined. The displacement may be calculated as follows:

Area of piston, 0.7854 $\times$ 15 $\times$ 15	=	176.715 sq. in.
½ area of piston rod, ½ $\times$ 0.7854 $\times$ 3 $\times$ 3	=	3.534 sq. in.
Average area of piston	=	173.181 sq. in.
Displacement per revolution .... 173.181 $\times$ 30 $\times$ 2	=	10391 cu. in.
Displacement per min. .... 10391 $\times$ 70	=	727370 cu. in.

Density.—The density of a body is the mass per unit volume. It may be found by dividing the mass of a body by its volume.

Specific Volume.—The specific volume of a substance is defined as the volume of a unit mass of the substance. It may be found by dividing the volume of a quantity of substance by the mass.

Velocity.—Velocity is the rate of motion of a body and is measured by the distance passed over in a unit of time. Velocity is generally expressed in ft. per sec. It may also be expressed in ft. per min. or miles per hour. Velocity is found by dividing the space by the time in seconds.

Work and Mechanical Energy.—Energy may be defined as the capacity or ability to do work, while work may be said to consist of changing the state of motion or the state of stress of matter, in opposition to forces which tend to resist such changes. The amount of work accomplished is determined by the magnitude of the force and the distance through which the force acts. The work done is found by multiplying the force by the distance.

The unit of work is then the foot-pound, which is the quantity of work or energy expended when a force of one pound acts through a distance of one foot in the direction of the force.

If, in Example 2, the compressor is assumed to operate under such conditions of high and low pressure that the average resistance is 72 lbs. per sq. in. of piston area, the work performed in one stroke may be calculated as follows:

Total force on piston....................	$173.18 \times 72 = 12468.96$ lbs.
Length of stroke.........................	$30 \div 12 = 2.5$ ft.
Work	$12468.96 \times 2.5 = 31172.4$ ft.-lbs.

Power.—The rate of performing work is called power. Thus power involves the idea of the time rate of performing work and may be found by dividing the work in foot pounds by the time in seconds.

The unit of power is the horsepower which is the performance of work at the rate of 33,000 ft.-lbs. p.m. or 550 ft.-lbs. p.s. The horsepower may be found by dividing the product of the force and its travel in one minute by 33,000.

Thus the horsepower of the compressor in the preceding example may be calculated as follows:

Force on piston.......................	$=12468.96$ lbs.
Piston travel per min.	$=2 \times 70 \times 2.5 = 350$ ft.
Horsepower	$=12468.96 \times 350 \div 33000 = 132.2$

Pressure.—Intensity of pressure, which for brevity is called pressure, is the rate of application of a uniformly distributed force upon an area. It is found by dividing the total force by the total area. The unit of pressure is generally expressed as pounds per square inch. In ordinary engineering work it is customary to measure pressure by means of gauges which really indicate only the excess of pressure above that of the atmosphere. The pressure recorded by such an instrument is termed gauge pressure, and to find the absolute or true pressure, the pressure of the atmosphere as determined by means of a barometer

must be added to the gauge pressure. Similarly gauges that indicate pressure below atmospheric pressure are known as vacuum or draft gauges. If these instruments indicate pressures below that of the atmosphere, then these pressures must be subtracted from that of the atmosphere in order to find the absolute or true pressures.

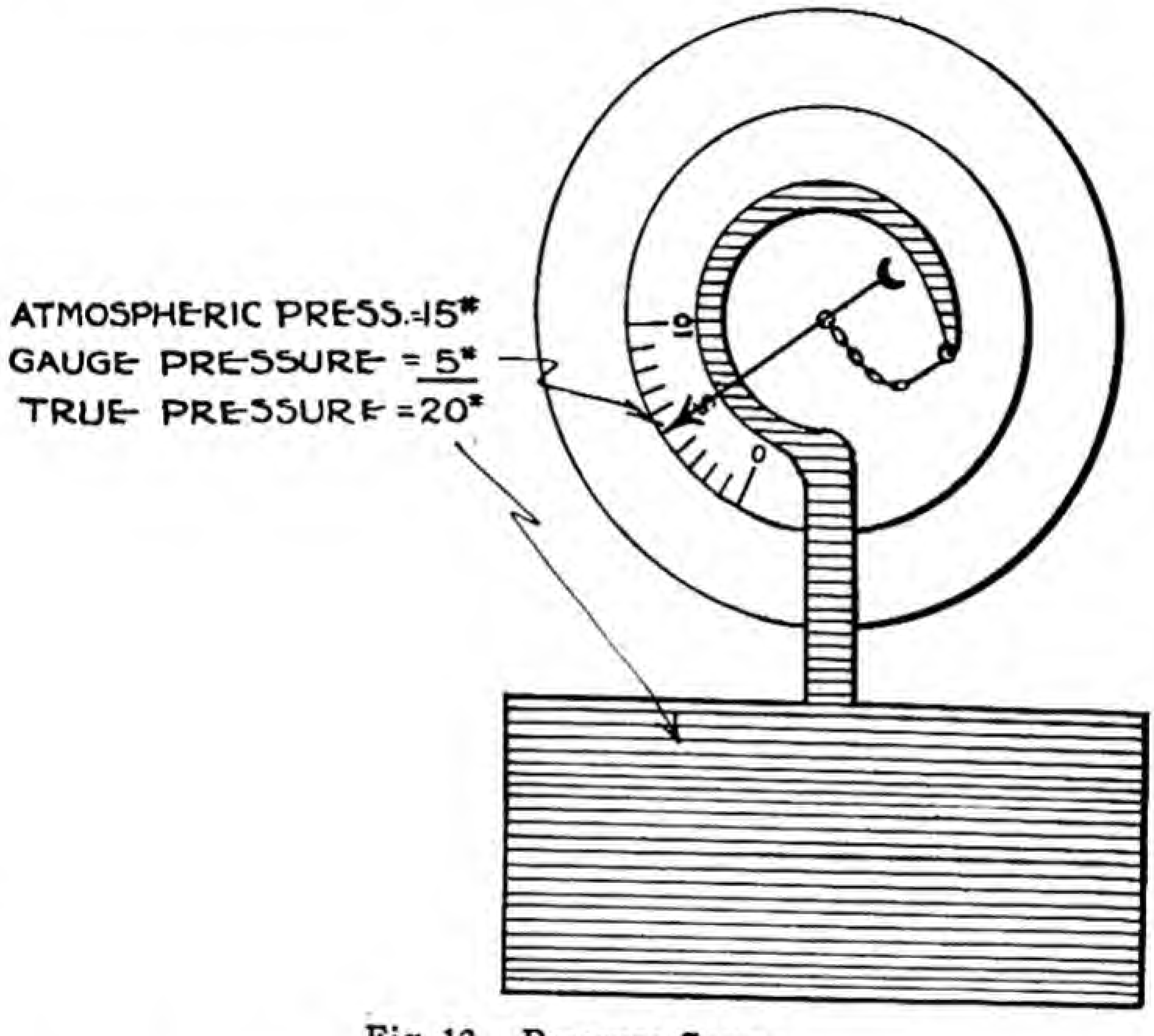


Fig. 12.—Pressure Gauge.

Pressure Gauges.—The most common type of gauge is the round indicating type, as shown by Fig. 12. This type of gauge consists of a flexible hollow tube of oval cross-section, bent into an arc of a circle. When the hollow oval tube is connected to a region of pressure higher than the atmosphere, the tube tends to straighten out, which movement is transmitted to the indicating hand by means of links, gears and pinions. A graduated dial, shown in Fig. 12, is placed just behind the pointer, so that the pressure may be read in suitable units.

Barometers.—The pressure of the atmosphere is generally measured by a mercury barometer. This consists of a glass tube about three feet long which is closed at the one end. After being filled with mercury, it is inverted in a bath of mercury, as shown by Fig. 13. The pressure of the atmosphere on the surface of the mercury bath sustains

a column of mercury about 29.9 in. high in the tube at sea level. The height of the column of mercury in inches multiplied by 0.491 will give the equivalent pressure in pounds per sq. in.

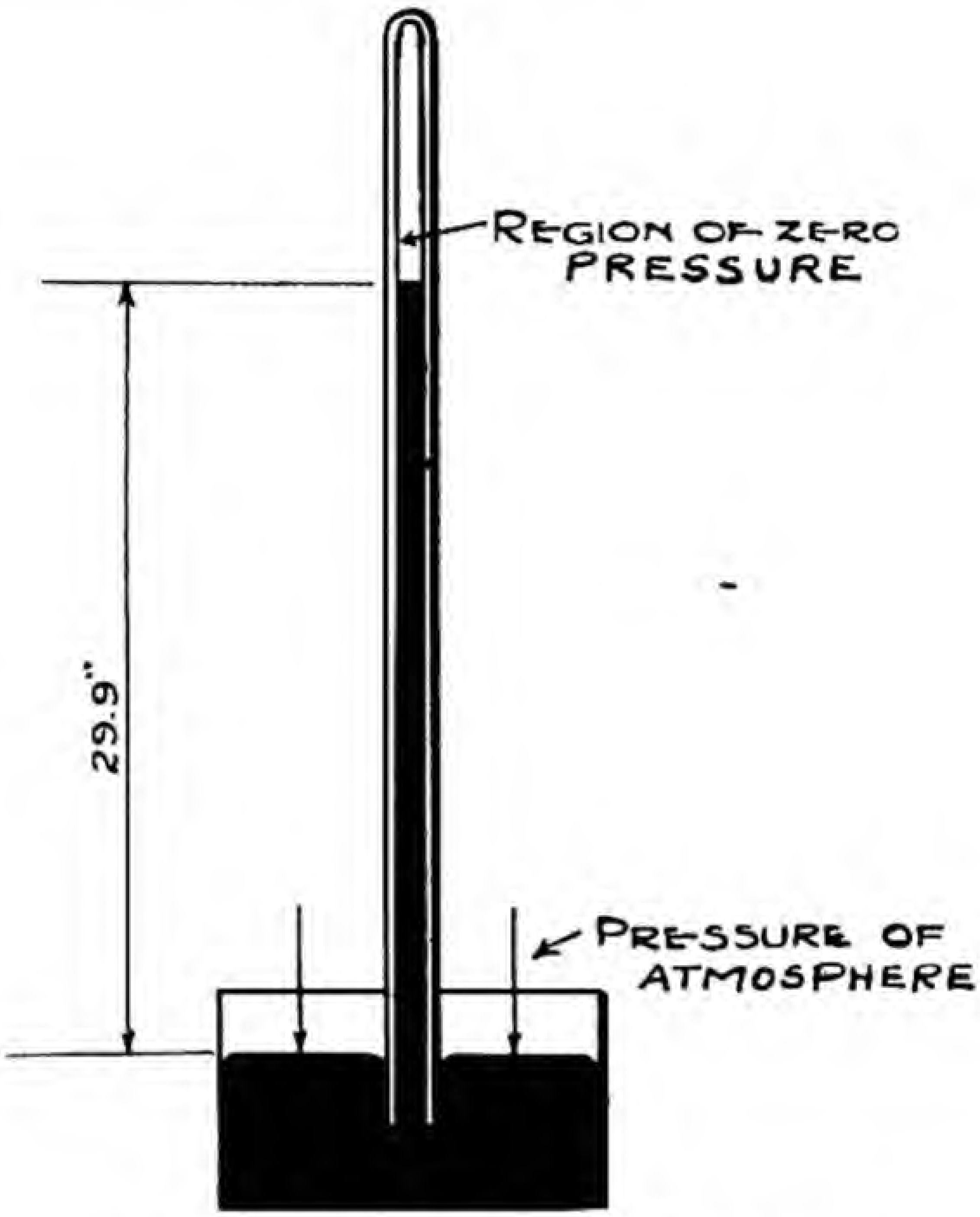


Fig. 13.—Simple Mercury Barometer.

Effects of Heat.—Since refrigeration deals with the transference of heat, it is important that we understand the effects resulting from the addition or subtraction of heat upon the thermal properties of matter. All matter is assumed to be made up of infinitely small groups of particles called molecules, which are further subdivided into atoms. These small particles are further assumed to be in constant motion, vibrating to and fro within very small orbits. The addition of heat to any substance will cause an increase in the vibration of its molecules, resulting in an increase in the total (energy) heat content.

This increase in energy of a substance while in its solid, liquid or gaseous state will cause an increase in temperature and volume at constant pressure; or an increase in temperature and pressure at constant volume. During a change of state at constant pressure in any sub-

stance from a solid to a liquid, or from a liquid to a gas, its temperature and volume will increase; or, if the volume is held constant, an increase in pressure and temperature will result. During the extraction of heat in any of the above processes the effects obtained will be reversed.

If a sufficient quantity of heat is added to a substance it is possible to convert it from one state to another as per example, from ice to water; from water to wet steam; from wet steam to dry and saturated steam; and from dry and saturated steam to superheated steam. Known the state or condition of the substance at any point its total heat content or heat energy may be determined from or computed from the table of the properties of that substance, as found in Chapter III.

Temperature.—The temperature of a body or substance denotes the degree or the intensity of heat. Bodies not passing through a change of state, but having different rates of motion of molecules have different (temperatures) intensities of heat. When there is no vibration of the molecules, there is no heat in the substance, and this point is known as the absolute zero of temperature. This point is 459.6° below 0° on the Fahrenheit scale and 273.1° on the Centigrade scale. Temperature measures intensity of heat and not quantity of heat.

Measurement of Temperature.—The intensity of heat is measured by thermometers. In ordinary refrigeration work, mercury thermometers are employed. These depend upon the expansion and contraction of mercury to indicate the changes of temperature. The unit of temperature measurement is called the degree. Reference points such as the freezing and boiling points of water have been taken as bases, and the expansion and contraction of the mercury is noted on a scale between the base points. The space between the base points is divided into equal divisions, or degrees. In the Fahrenheit scale there are 180 degrees or divisions between the two reference points and the freezing point is placed at 32 and the boiling point at 212, while on the Centigrade scale there are 100 degrees between reference points, and the freezing point is at 0 and the boiling point is at 100. The Fahrenheit scale was invented by Gabriel Daniel Fahrenheit, of Germany. The Centigrade, or Celsius scale, was invented by Anders Celsius, a Swedish astronomer. The word centigrade is derived from the Latin words, *centum*, meaning a hundred, and *gradus* meaning degree. The Fahrenheit thermometer scale of temperature is used almost exclusively in the United States in refrigeration work. The relationship of the two thermometer scales is shown by Fig. 14. It will be noted that the scales read the same at one point only, and that this is at −40°.

When it is desired, readings on the Centigrade scale may be transferred to Fahrenheit degrees, since 1° C. is equivalent to 1.8° F. The conversion is shown by the following equation:

TABLE 1.—TEMPERATURE CONVERSIONS, FAHRENHEIT AND CENTIGRADE.

F.	C.	F.	C.	F.	C.	F.	C.	F.	C.	F.	C.	F.	C.
−40	−40.	26	−3.3	92	33.3	158	70.	224	106.7	290	143.3	360	182.2
−39	−39.4	27	−2.8	93	33.9	159	70.6	225	107.2	291	143.9	370	187.8
−38	−38.9	28	−2.2	94	34.4	160	71.1	226	107.8	292	144.4	380	193.3
−37	−38.3	29	−1.7	95	35.	161	71.7	227	108.3	293	145.	390	198.9
−36	−37.8	30	−1.1	96	35.6	162	72.2	228	108.9	294	145.6	400	204.4
−35	−37.2	31	−0.6	97	36.1	163	72.8	229	109.4	295	146.1	410	210.
−34	−36.7	32	0.	98	36.7	164	73.3	230	110.	296	146.7	420	215.6
−33	−36.1	33	+0.6	99	37.2	165	73.9	231	110.6	297	147.2	430	221.1
−32	−35.6	34	1.1	100	37.8	166	74.4	232	111.1	298	147.8	440	226.7
−31	−35.	35	1.7	101	38.3	167	75.	233	111.7	299	148.3	450	232.2
−30	−34.4	36	2.2	102	38.9	168	75.6	234	112.2	300	148.9	460	237.8
−29	−33.9	37	2.8	103	39.4	169	76.1	235	112.8	301	149.4	470	243.3
−28	−33.3	38	3.3	104	40.	170	76.7	236	113.3	302	150.	480	248.9
−27	−32.8	39	3.9	105	40.6	171	77.2	237	113.9	303	150.6	490	254.4
−26	−32.2	40	4.4	106	41.1	172	77.8	238	114.4	304	151.1	500	260.
−25	−31.7	41	5.	107	41.7	173	78.3	239	115.	305	151.7	510	265.6
−24	−31.1	42	5.6	108	42.2	174	78.9	240	115.6	306	152.2	520	271.1
−23	−30.6	43	6.1	109	42.8	175	79.4	241	116.1	307	152.8	530	276.7
−22	−30.	44	6.7	110	43.3	176	80.	242	116.7	308	153.3	540	282.2
−21	−29.4	45	7.2	111	43.9	177	80.6	243	117.2	309	153.9	550	287.8
−20	−28.9	46	7.8	112	44.4	178	81.1	244	117.8	310	154.4	560	293.3
−19	−28.3	47	8.3	113	45.	179	81.7	245	118.3	311	155.	570	298.9
−18	−27.8	48	8.9	114	45.6	180	82.2	246	118.9	312	155.6	580	304.4
−17	−27.2	49	9.4	115	46.1	181	82.8	247	119.4	313	156.1	590	310.
−16	−26.7	50	10.	116	46.7	182	83.3	248	120.	314	156.7	600	315.6
−15	−26.1	51	10.6	117	47.2	183	83.9	249	120.6	315	157.2	610	321.1
−14	−25.6	52	11.1	118	47.8	184	84.4	250	121.1	316	157.8	620	326.7
−13	−25.	53	11.7	119	48.3	185	85.	251	121.7	317	158.3	630	332.2
−12	−24.4	54	12.2	120	48.9	186	85.6	252	122.2	318	158.9	640	337.8
−11	−23.9	55	12.8	121	49.4	187	86.1	253	122.8	319	159.4	650	343.3
−10	−23.3	56	13.3	122	50.	188	86.7	254	123.3	320	160.	660	348.9
−9	−22.8	57	13.9	123	50.6	189	87.2	255	123.9	321	160.6	670	354.4
−8	−22.2	58	14.4	124	51.1	190	87.8	256	124.4	322	161.1	680	360.
−7	−21.7	59	15.	125	51.7	191	88.3	257	125.	323	161.7	690	365.6
−6	−21.1	60	15.6	126	52.2	192	88.9	258	125.6	324	162.2	700	371.1
−5	−20.6	61	16.1	127	52.8	193	89.4	259	126.1	325	162.8	710	376.7
−4	−20.	62	16.7	128	53.3	194	90.	260	126.7	326	163.3	720	382.2
−3	−19.4	63	17.2	129	53.9	195	90.6	261	127.2	327	163.9	730	387.8
−2	−18.9	64	17.8	130	54.4	196	91.1	262	127.8	328	164.4	740	393.3
−1	−18.3	65	18.3	131	55.	197	91.7	263	128.3	329	165.	750	[illegible]
0	−17.8	66	18.9	132	55.6	198	92.2	264	128.9	330	165.6	760	404.4
+1	−17.2	67	19.4	133	56.1	199	92.8	265	129.4	331	166.1	770	410
2	−16.7	68	20.	134	56.7	200	93.3	266	130.	332	166.7	780	415.6
3	−16.1	69	20.6	135	57.2	201	93.9	267	130.6	333	167.2	790	421.1
4	−15.6	70	21.1	136	57.8	202	94.4	268	131.1	334	167.8	800	426.7
5	−15.	71	21.7	137	58.3	203	95.	269	131.7	335	168.3	810	432.2
6	−14.4	72	22.2	138	58.9	204	95.6	270	132.2	336	168.9	820	437.8
7	−13.9	73	22.8	139	59.4	205	96.1	271	132.8	337	169.4	830	443.3
8	−13.3	74	23.3	140	60.	206	96.7	272	133.3	338	170.	840	448.9
9	−12.8	75	23.9	141	60.6	207	97.2	273	133.9	339	170.6	850	454.4
10	−12.2	76	24.4	142	61.1	208	97.8	274	134.4	340	171.1	860	460.
11	−11.7	77	25.	143	61.7	209	98.3	275	135.	341	171.7	870	465.6
12	−11.1	78	25.6	144	62.2	210	98.9	276	135.6	342	172.2	880	471.1
13	−10.6	79	26.1	145	62.8	211	99.4	277	136.1	343	172.8	890	476.7
14	−10.	80	26.7	146	63.3	212	100.	278	136.7	344	173.3	900	482.2
15	− 9.4	81	27.2	147	63.9	213	100.6	279	137.2	345	173.9	910	487.8
16	− 8.9	82	27.8	148	64.4	214	101.1	280	137.8	346	174.4	920	493.3
17	− 8.3	83	28.3	149	65.	215	101.7	281	138.3	347	175.	930	498.9
18	− 7.8	84	28.9	150	65.6	216	102.2	282	138.9	348	175.6	940	504.4
19	− 7.2	85	29.4	151	66.1	217	102.8	283	139.4	349	176.1	950	510.
20	− 6.7	86	30.	152	66.7	218	103.3	284	140.	350	176.7	960	515.6
21	− 6.1	87	30.6	153	67.2	219	103.9	285	140.6	351	177.2	970	521.1
22	− 5.6	88	31.1	154	67.8	220	104.4	286	141.1	352	177.8	980	526.7
23	− 5.	89	31.7	155	68.3	221	105.	287	141.7	353	178.3	990	532.2
24	− 4.4	90	32.2	156	68.9	222	105.6	288	142.2	354	178.9	1000	537.8
25	− 3.9	91	32.8	157	69.4	223	106.1	289	142.8	355	179.4	1010	543.3

TABLE 2.—TEMPERATURE CONVERSIONS, CENTIGRADE AND FAHRENHEIT.

C.	F.	C.	F.	C.	F.	C.	F.	C.	F.	C.	F.	C.	F.
−40	−40.	26	78.8	92	197.6	158	316.4	224	435.2	290	554	950	1742
−39	−38.2	27	80.6	93	199.4	159	318.2	225	437.	300	572	960	1760
−38	−36.4	28	82.4	94	201.2	160	320.	226	438.8	310	590	970	1778
−37	−34.6	29	84.2	95	203.	161	321.8	227	440.6	320	608	980	1796
−36	−32.8	30	86.	96	204.8	162	323.6	228	442.4	330	626	990	1814
−35	−31.	31	87.8	97	206.6	163	325.4	229	444.2	340	644	1000	1832
−34	−29.2	32	89.6	98	208.4	164	327.2	230	446.	350	662	1010	1850
−33	−27.4	33	91.4	99	210.2	165	329.	231	447.8	360	680	1020	1868
−32	−25.6	34	93.2	100	212.	166	330.8	232	449.6	370	698	1030	1886
−31	−23.8	35	95.	101	213.8	167	332.6	233	451.4	380	716	1040	1904
−30	−22.	36	96.8	102	215.6	168	334.4	234	453.2	390	734	1050	1922
−29	−20.2	37	98.6	103	217.4	169	336.2	235	455.	400	752	1060	1940
−28	−18.4	38	100.4	104	219.2	170	338.	236	456.8	410	770	1070	1958
−27	−16.6	39	102.2	105	221.	171	339.8	237	458.6	420	788	1080	1976
−26	−14.8	40	104.	106	222.8	172	341.6	238	460.4	430	806	1090	1994
−25	−13.	41	105.8	107	224.6	173	343.4	239	462.2	440	824	1100	2012
−24	−11.2	42	107.6	108	226.4	174	345.2	240	464.	450	842	1110	2030
−23	− 9.4	43	109.4	109	228.2	175	347.	241	465.8	460	860	1120	2048
−22	− 7.6	44	111.2	110	230.	176	348.8	242	467.6	470	878	1130	2066
−21	− 5.8	45	113.	111	231.8	177	350.6	243	469.4	480	896	1140	2084
−20	− 4.	46	114.8	112	233.6	178	352.4	244	471.2	490	914	1150	2102
−19	− 2.2	47	116.6	113	235.4	179	354.2	245	473.	500	932	1160	2120
−18	− 0.4	48	118.4	114	237.2	180	356.	246	474.8	510	950	1170	2138
−17	+ 1.4	49	120.2	115	239.	181	357.8	247	476.6	520	968	1180	2156
−16	3.2	50	122.	116	240.8	182	359.6	248	478.4	530	986	1190	2174
−15	5.	51	123.8	117	242.6	183	361.4	249	480.2	540	1004	1200	2192
−14	6.8	52	125.6	118	244.4	184	363.2	250	482.	550	1022	1210	2210
−13	8.6	53	127.4	119	246.2	185	365.	251	483.8	560	1040	1220	2228
−12	10.4	54	129.2	120	248.	186	366.8	252	485.6	570	1058	1230	2246
−11	12.2	55	131.	121	249.8	187	368.6	253	487.4	580	1076	1240	2264
−10	14.	56	132.8	122	251.6	188	370.4	254	489.2	590	1094	1250	2282
− 9	15.8	57	134.6	123	253.4	189	372.2	255	491.	600	1112	1260	2300
− 8	17.6	58	136.4	124	255.2	190	374.	256	492.8	610	1130	1270	2318
− 7	19.4	59	138.2	125	257.	191	375.8	257	494.6	620	1148	1280	2336
− 6	21.2	60	140.	126	258.8	192	377.6	258	496.4	630	1166	1290	2354
− 5	23.	61	141.8	127	260.6	193	379.4	259	498.2	640	1184	1300	2372
− 4	24.8	62	143.6	128	262.4	194	381.2	260	500.	650	1202	1310	2390
− 3	26.6	63	145.4	129	264.2	195	383.	261	501.8	660	1220	1320	2408
− 2	28.4	64	147.2	130	266.	196	384.8	262	503.6	670	1238	1330	2426
− 1	30.2	65	149.	131	267.8	197	386.6	263	505.4	680	1256	1340	2444
0	32.	66	150.8	132	269.6	198	388.4	264	507.2	690	1274	1350	2462
+ 1	33.8	67	152.6	133	271.4	199	390.2	265	509.	700	1292	1360	2480
2	35.6	68	154.4	134	273.2	200	392.	266	510.8	710	1310	1370	2498
3	37.4	69	156.2	135	275.	201	393.8	267	512.6	720	1328	1380	2516
4	39.2	70	158.	136	276.8	202	395.6	268	514.4	730	1346	1390	2534
5	41.	71	159.8	137	278.6	203	397.4	269	516.2	740	1364	1400	2552
6	42.8	72	161.6	138	280.4	204	399.2	270	518.	750	1382	1410	2570
7	44.6	73	163.4	139	282.2	205	401.	271	519.8	760	1400	1420	2588
8	46.4	74	165.2	140	284.	206	402.8	272	521.6	770	1418	1430	2606
9	48.2	75	167.	141	285.8	207	404.6	273	523.4	780	1436	1440	2624
10	50.	76	168.8	142	287.6	208	406.4	274	525.2	790	1454	1450	2642
11	51.8	77	170.6	143	289.4	209	408.2	275	527.	800	1472	1460	2660
12	53.6	78	172.4	144	291.2	210	410.	276	528.8	810	1490	1470	2678
13	55.4	79	174.2	145	293.	211	411.8	277	530.6	820	1508	1480	2696
14	57.2	80	176.	146	294.8	212	413.6	278	532.4	830	1526	1490	2714
15	59.	81	177.8	147	296.6	213	415.4	279	534.2	840	1544	1500	2732
16	60.8	82	179.6	148	298.4	214	417.2	280	536.	850	1562	1510	2750
17	62.6	83	181.4	149	300.2	215	419.	281	537.8	860	1580	1520	2768
18	64.4	84	183.2	150	302.	216	420.8	282	539.6	870	1598	1530	2786
19	66.2	85	185.	151	303.8	217	422.6	283	541.4	880	1616	1540	2804
20	68.	86	186.8	152	305.6	218	424.4	284	543.2	890	1634	1550	2822
21	69.8	87	188.6	153	307.4	219	426.2	285	545.	900	1652	1600	2912
22	71.6	88	190.4	154	309.2	220	428.	286	546.8	910	1670	1650	3002
23	73.4	89	192.2	155	311.	221	429.8	287	548.6	920	1688	1700	3092
24	75.2	90	194.	156	312.8	222	431.6	288	550.4	930	1706	1750	3182
25	77.	91	195.8	157	314.6	223	433.4	289	552.2	940	1724	1800	3272

Fahrenheit degrees = 1.8 × Centigrade degrees + 32°.

Thus if the temperature of a cold storage room is 2° C., the equivalent Fahrenheit temperature is found as follows:

Fahrenheit temperature = 1.8 × 2 + 32 = 35.6° F.

These temperature conversions may also be read directly from Tables 1 and 2.

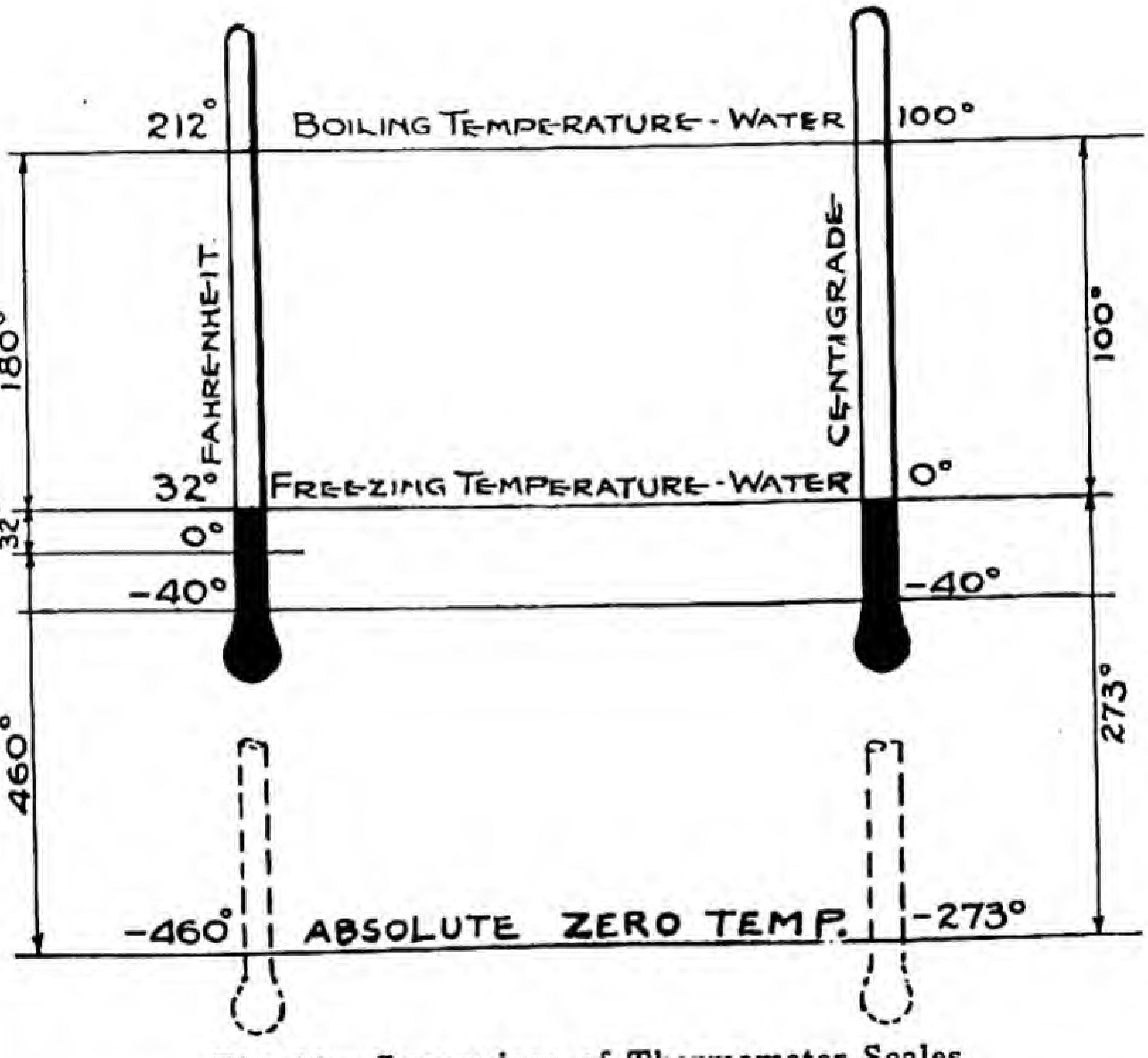


Fig. 14.—Comparison of Thermometer Scales.

The readings on the Fahrenheit and Centigrade scales may be reduced readily to the absolute scale. It is only necessary to add 459.6° to the scale readings of the Fahrenheit thermometer to convert it into its equivalent on the absolute scale, and likewise, 273.1° may be added to the Centigrade reading to secure the absolute Centigrade temperature, thus:

Abs. F. Temp. = Degrees, F. + 459.6° and
Abs. C. Temp. = Degrees, C. + 273.1°

Heat Quantity and British Thermal Unit.—It is necessary in engineering computations involving the addition or extraction of heat to

have a measure of heat quantity. The British thermal unit (Btu.) has been adopted by English speaking countries as the standard unit of measurement. It is the amount of heat required to increase or lower the temperature of one pound of water one degree Fahrenheit, at its maximum density, at which point its specific heat is unity. For all practical purposes the specific heat of water between 32° F. and 212° F. may be taken as unity.

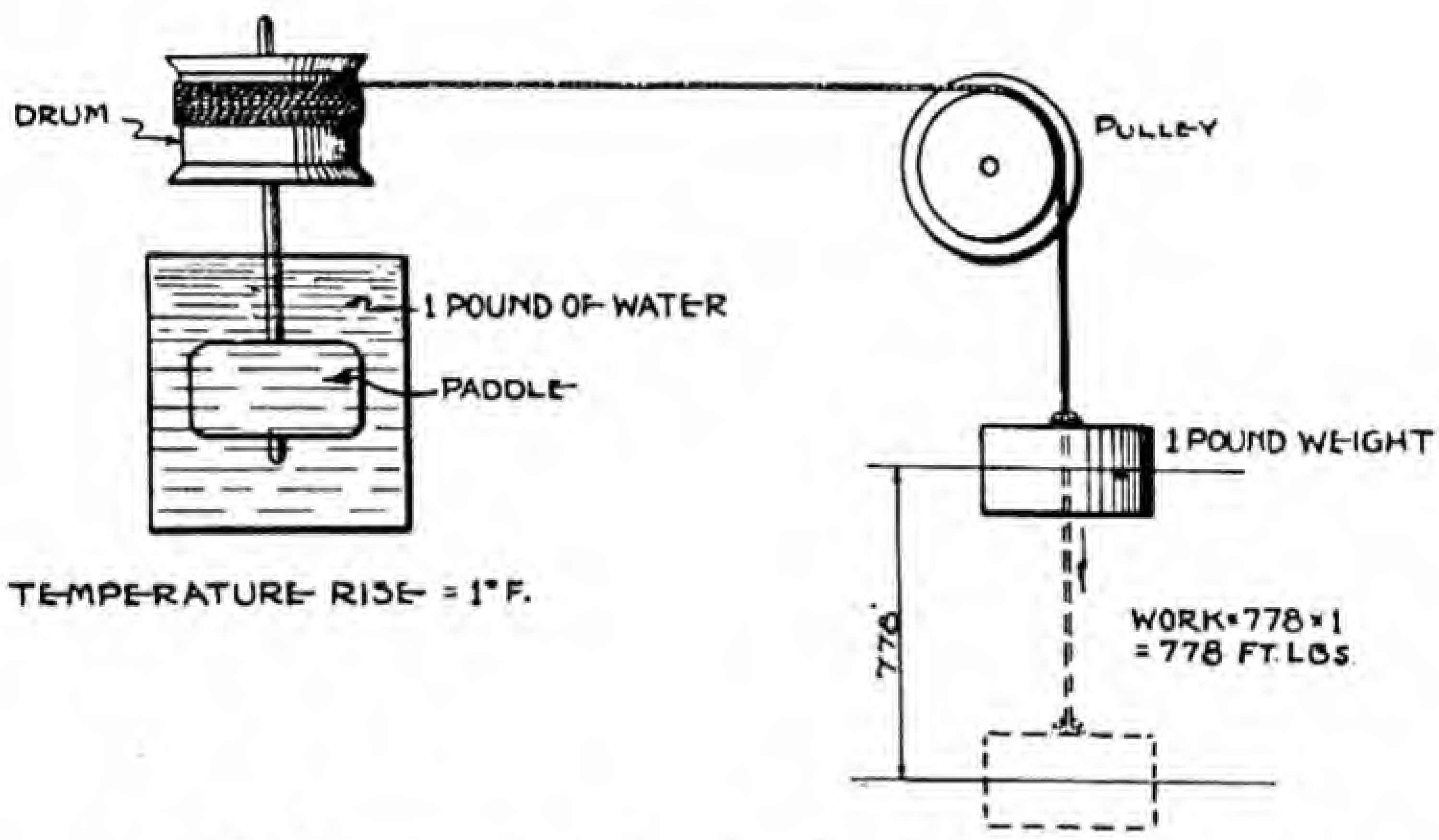


Fig. 15.—Apparatus to Determine Relation of Heat and Work.

The quantity of heat added to or subtracted from a substance may be computed by multiplying its weight in pounds by its specific heat and its temperature change in degrees Fahrenheit. The above holds true only when there is no change in state of the substance.

Mechanical Equivalent of Heat.—One of the fundamental laws of physical science states that the total energy of the universe remains constant and that the energy can not be increased, diminished, created or destroyed. From this law it is also known that the different forms of energy are mutually interconvertible. Thus mechanical energy (or work) and heat are interconvertible. From many experiments by many men at different times and places, it has been shown that 778 ft-lbs. of work may be transformed into one Btu. The number, 778 ft-lbs., is termed the mechanical equivalent of heat. Fig. 15 shows a means of determining the relation of heat and work. The pound weight in being lowered 778 feet produces 778-ft-lbs. of work, which are transferred without friction to the water by means of the rope and pulleys. This work heats the pound of water 1° F.

Specific Heat.—If a small quantity of heat is added to or extracted from a substance, without changing its physical or chemical state, a small change of temperature is produced.

If the weight of the substance was 1 lb. and the temperature change 1° F. then the quantity of heat necessary to make the change in temperature is known as the specific heat. The specific heat of a substance may be defined as the quantity of heat in Btu. required to raise or lower 1 lb. of the substance 1° F. Thus it will be seen that the specific heat of a substance represents its capacity for absorbing heat when compared with water. Specific heats of some common substances are shown in Table 3.

Sensible and Latent Heat.—When heat is added to or extracted from a substance without a change of state, the temperature is increased or decreased. The heat thus added or extracted is known as sensible heat, since the heat transfer is perceived by a change of temperature. Similarly, when heat is added to or extracted from a substance without a change of temperature, a change of state is produced, and the heat so added or extracted is known as the latent heat.

Heat Content or Total Heat.—The heat content of a body or substance is the total quantity of heat in the substance above a given reference temperature such as 32° F. or −40° F. Thus the heat content of liquid carbon dioxide at 70° F. is 24.78 Btu. per lb. above 32° F. as a reference plane, while the total heat of saturated ammonia vapor at 80° F. is 630.7 Btu. above −40° F.

Entropy.—It is convenient in solving certain problems to make use of a ratio or abstract quantity which is called entropy. For all practical purposes the entropy of a substance at a given state is the mathematical ratio obtained by dividing the amount of heat necessary to bring the substance to the given state by the average absolute temperature during the heat transfer. It must be noted that entropy is a mere ratio and has no physical existence. Thus, ammonia changing from liquid to vapor state at 70° F. may be considered. The absolute temperature is equal to $459.6 + 70 = 529.6°$. The latent heat of evaporation is 508.6 Btu. Therefore the entropy of evaporation is equal to $508.6 \div 529.6 = 0.960$.

Fusion.—When a substance changes from the solid to the liquid state or from the liquid to the solid state, the heat added or extracted is known as the latent heat of fusion. Thus when ice melts in a refrigerator the addition of 144 Btu. per lb. is required to produce the melting, or when ice freezes in an ice tank 144 Btu. per lb. must be extracted.

TABLE 3.—PROPERTIES OF VARIOUS SUBSTANCES.

Name of substance.	Specific gravity Water = 1.000	Temperature or Temp. range for Sp. Ht. in degrees F.	Specific heat Btu. per lb.	Melting Temp. in degrees F.	Heat of fusion Btu. per lb.	Boiling Temp. in degrees F.	Heat of evaporation Btu. per lb.
Acetic Acid	1.052	78.8–204.8	0.5220			246.2	152.3
Acetone	0.797		0.5150			132.8	225.7
Air	0.08073*		0.2400			−314	79.5
Alcohol (ethyl)	0.785	−4	0.5053	−202		172.4	
Alcohol (methyl)	0.790	53.6	0.600			152.6	470
Alumina		32 −212	0.1827				
Aluminum (cast)	2.56	68 −212	0.2145	1217	432		
Amyl Alcohol	0.817					278.6	216
Ammonia Nitrate				142–154	113		
Aniline	1.028	46.4–179.6	0.512	17.6		359.6	187.7
Anthracine	1.147			415.4		680	
Anthraquinone	1.438			524.4		716	
Antimony	6.71	32 −212	0.495				
Arsenic Trioxide		55.4–206.6	0.1276				
Asbestos		68 −212	0.20				
Barium Nitrate		55.4–208.4	0.1523	1058		Decom	
Benzene	0.879	50 −104	0.3402			176	
Benzoic Acid	1.266			249.8		480.2	
Benzol		42.8–140	0.4194				167.1
Benzol, solid		−22	0.3130				
Berylium		32 −212	0.4246				
Bismuth	9.75	71.6–212	0.3035		22.7		
Bismuth, fluid		536 −716	0.363				
Brass			0.090				
Brick work			0.20				
Bromine	3.15				29.1		
Bronze		59 −208	0.0858	1690			
Cadmium	8.66	32 −212	0.0548		24.6		
Caesium	1.88						
Calcium	1.58	32 −212	0.1804				
Calcium Carbonate		32 −572	0.2204				
Calcium Chloride					64.3		
Calcium Fluoride		59 −212.2	0.2154	2372			
Calcium Sulphate		32 −572	0.1908	2480			
Camphor	9.992			247		401	
Carbon Disulphide		−22	0.1569			+115	156
Carbon Tetrachloride	1.608	32	0.1980	−11.2		170.6	83.3
Carbonic Oxide			0.245			−310	
Cellulose	1.36		0.37				
Charcoal			0.2415				
Chlorine, vapor	0.21		0.1210			−28	
Chlorine, liquid	1.44					−28.5	121
Chloroform	1.48	59 −95	0.2337			141.8	105.2
Chromium	6.50		0.1039				
Citric Acid	1.542			309.2			
Clay		68 −208.4	0.2243				
Coal			0.2411				
Cobalt	8.65	59 −212	0.1030				
Coke			0.2008				

*Weight per cu. ft. at 32 degrees F.

TABLE 3.—PROPERTIES OF VARIOUS SUBSTANCES.—Continued.

Name of substance.	Specific gravity Water=1.000	Temperature or Temp. range for Sp. Ht. in degrees F.	Specific heat Btu. per lb.	Melting Temp. in degrees F.	Heat of fusion Btu. per lb.	Boiling Temp. in degrees F.	Heat of evaporation Btu. per lb.
Copper, cast	8.60	32 -212	0.0933				
Copper Oxide		53.6-208.4	0.1420				
Copper Pyrites		66.2-118.4	0.131				
Creosote	1.111					426.2	
Crotonylene	0.14290*					64	
Decane		57.2- 64.4	0.5058				109.5
Ebonite		68 -212	0.40				
Ether	0.718	86	0.547				159
Ether, ethyl			0.5267				169
Ethyl Alcohol	0.794	32 -208.4	0.680			172.4	374
Ethyl Benzol (liquid)		32	0.3929				137.3
Ferric Oxide		75.2-210.2	0.1678	2822			
Flint Glass		50 -122	0.117				
Fluorspar		86	0.21				
Formaldehyde	0.815						
Formic Acid	1.245	32	0.4966	17.6		212	217
Gallium	5.95				34.4		
Gas Report Carbon			0.2038				
Gasoline		50 - 68	0.5350				
German Silver		32 -212	0.095				
Glass			0.16-0.18				
Glass Crown		50 -122	0.16				
Glycerine	1.260	59 -122	0.576			554	
Gold, cast	19.26	32 -212	0.0316				
Granite		68 -212	0.19				
Graphite			0.2018				
Gypsum		60.8-114.8	0.259				
Heptane		64.4-123.8	0.4869				140
Hexane		60.8- 98.6	0.5042			+156	147
Hydrochloric Acid			0.6000				
Hydrogen	0.00559*		3.41			−412	360
Hydrogen Sulphide	0.09012*		0.2423			− 82	
Ice	0.914		0.502	32	144		
India Rubber		86 -212	0.481				
Iridium	22.42	32 -212	0.0323				
Iron, pure	7.86	32	0.1050				
Wrought	7.86		0.1138				
Gray cast iron	7.10						
Steel	7.70						
White	7.65						
Isoamylene		−5.8- 57.2	0.4970				
Kerosene		50 - 68	0.4755				
Kryolite		60.8-131	0.253				
Lead	11.37	59	0.0299	617			
Lead Sulphate		32 -572	0.0478	2012			
Lead Sulphide			0.36	1850			
Leather, dry			0.2169				
Lime		59 -212	0.2166				
Limestone		66.2-122	0.553				

*Weight per cu. ft. at 32 degrees F.

TABLE 3.—PROPERTIES OF VARIOUS SUBSTANCES.—Continued.

Name of substance.	Specific gravity Water = 1.000	Temperature or Temp. range for Sp. Ht. in degrees F.	Specific heat Btu. per lb.	Melting Temp. in degrees F.	Heat of fusion Btu. per lb.	Boiling Temp. in degrees F.	Heat of evaporation Btu. per lb.
Litharge	....	64.4	0.21	....	....	....	....
Lithium	0.59	80.6–210.2	0.9408	....	....	....	....
Magnesium	1.74	32	0.2456	....	....	....	....
Malic Acid	1.601			212	....	....	....
Manganese	7.42	57.2–206.6	0.1217	....	....	....	....
Marble		68 –208.4	0.2100	....	....	....	....
Mercury	13.55	-108 to -40	0.0334	-39	5.1	....	122
Mesitylene	0.31755*	....		....	....	+326	....
Mica	....	50 – 68	0.314	....	....	....	....
Milk Sugar	1.525	....		....	....	....	....
Molybdenum	8.56	59 –824	0.0740	....	....	....	....
Naphthalene	1.152	....	0.445	176	64.2	424.4	....
Naphthol	1.224	....	....	201.2	....	536	....
Nickel	8.70	64.4–212	0.109	....	....	....	....
Nitric Acid	0.12303*	....	0.5700	....	....	186.8	207
Nitric Oxide	0.08383*	....	0.2317	....	....	–254	....
Nitrobenzene	1.187	....	....	37.4	....	401	....
Nitrogen (gas)	0.07845*	....	0.244	....	....	–318	....
Nitrogen, liquid	0.80	....	....	....	....	–318	86.4
Oak wood	....	44.6	0.47	....	....	....	....
Octane	....	53.6 –66.2	0.5111	....	....	....	128
Oils, Castor	....	....	0.434	....	....	....	....
Citron	0.818	40	0.438	....	....	....	....
Cottonseed	....	....	0.4702	....	....	....	....
Turpentine	....	32	0.4106	14	....	....	....
Olive Oil	0.911	32 – 68	0.69	....	....	....	....
Oxygen	0.08926*		0.217	....	....	–297	106.1
Paraffin	....	64.4–210.2	0.498	....	63.3	....	....
Paraffin	....	32 – 68	0.6369	....	....	....	....
Petroleum	....	70 –136	0.511	....	....	....	....
Phenol	1.060	....	....	107.6	44.9	357.8	....
Platinum, cast	21.50	64.4–212	0.0324	....	....	....	....
Portland Cement	....	....	0.246	....	....	....	....
Clinker	....	73.4–210.2	0.2163	....	....	....	....
Potassium	0.87	....	0.1662	....	28.3	....	....
Potassium							
Carbonate	....	60.8–208.4	0.2096	....	....	....	....
Chlorate	....	57.6–210.2	0.1730	....	....	....	....
Chloride	....	55.4–208.4	0.2388	....	....	....	....
Nitrate	....	59 –208.4	0.1901	....	85.2	....	....
Sulphate	....	59 –208.4	0.1357	....	....	....	....
Propyl Benzol	....	32	0.4000	....	....	....	129.2
Pyrites (iron)	....	32	0.174	....	....	....	....
Quartz	....	212	0.481	....	....	....	....
Quartz	....	32	0.1735	....	....	....	....
Rubber (para)	....	68 –208.4	0.1910	....	....	....	....
Sand	....		0.195	....	....	....	....
Sand (quartz)	....	68 –986	0.316	....	....	....	....
Sea Water	1.0235	64	0.938	....	....	....	....
Sea Water	1.0463	64	0.903	....	....	....	....

Weight per cu. ft. at 32° F.

TABLE 3.—PROPERTIES OF VARIOUS SUBSTANCES.—Continued.

Name of substance.	Specific gravity Water=1.000	Temperature or Temp. range for Sp. Ht. in degrees F.	Specific heat Btu. per lb.	Melting Temp. in degrees F.	Heat of fusion Btu. per lb.	Boiling Temp. in degrees F.	Heat of evaporation Btu. per lb.
Sea Water	1.0043	64	0.98				
Silica		60.8–208.4	0.2728				
Silver, cast	10.53	32 –212	0.0559				
Sodium	0.98		0.2830				
Sodium Carbonate		59 –208.4	0.2140				
Sodium Chloride		57.2–208.4	0.2782				
Sodium Nitrate		62.6–208.4	0.2312				
Sodium Sulphate			0.200				
Spermaceti				120	66.6		
Steel			0.1170				
Stone		71.6–123.8	0.3005				
Sugar (cane)	1.588	59 –206.6	0.1764				
Sulphur		41 – 71.6	0.332		16.9		652
Sulphur— Amorphous	2.00	63 –113	0.163	239	17.0		
Sulphuric Acid	1.84	53.6–210.2	0.4400				220
Sulphurous Acid	0.17870*		0.154			+15	
Tallow				92			
Tetradecane		57.2– 69.8	0.4995				
Tin, cast	7.29	32 –212	0.0545	451			
Toluol		64.4	0.42				150.2
Tungsten	18.77	32 –212	0.0336				
Turpentine	0.859	68 –212	0.33				133.2
Turpentine, vapor		354.2–480.2	0.506				
Uranium	18.70	32 –208.4	0.0280				
Vanadium	5.50	32 –212	0.1153				
Vegetable Oil		32 – 50	0.40				
Vinegar			0.920				
Vulcanite			0.2415				
Water			1.00	32	144	212	970
Wax				142–154	76.1		
Wood		62.6–208.4	0.1248				
Wood Charcoal			0.45–0.65				
Xylol		32	0.3834				149.0
Zinc	7.14	32 –212	0.0935	787			
Zinc Oxide		32 –212	0.1146				
Zinc Sulphide		68 –210.2	0.872				

*Weight per cu. ft. at 32° F.

Evaporation and Condensation.—When a substance changes from liquid to vapor state, the heat required to produce the change is known as the latent heat of evaporation, and while in changing from vapor to liquid it is known as the latent heat of condensation. Thus the latent heat of condensation or evaporation of ammonia at 50° F. and a pressure of 89.19 lbs. absolute is 527.3 Btu. per lb.

Fig. 16 shows the absorption of the latent heat of evaporation of ammonia from the air in a cold storage room, 569.3 Btu. being required per pound of ammonia evaporated at 30 lbs. absolute pressure. The ammonia enters the container as a liquid and leaves in the form of a saturated vapor.

Table 3 gives some properties of various substances.

Absorption and Dissociation.—When certain substances are dissolved or absorbed by other substances, energy is required to change the conditions of the molecules, to suit the new state of the substance. The heat so added or extracted to produce the change is known as the latent heat of absorption or dissociation, as the case may be.

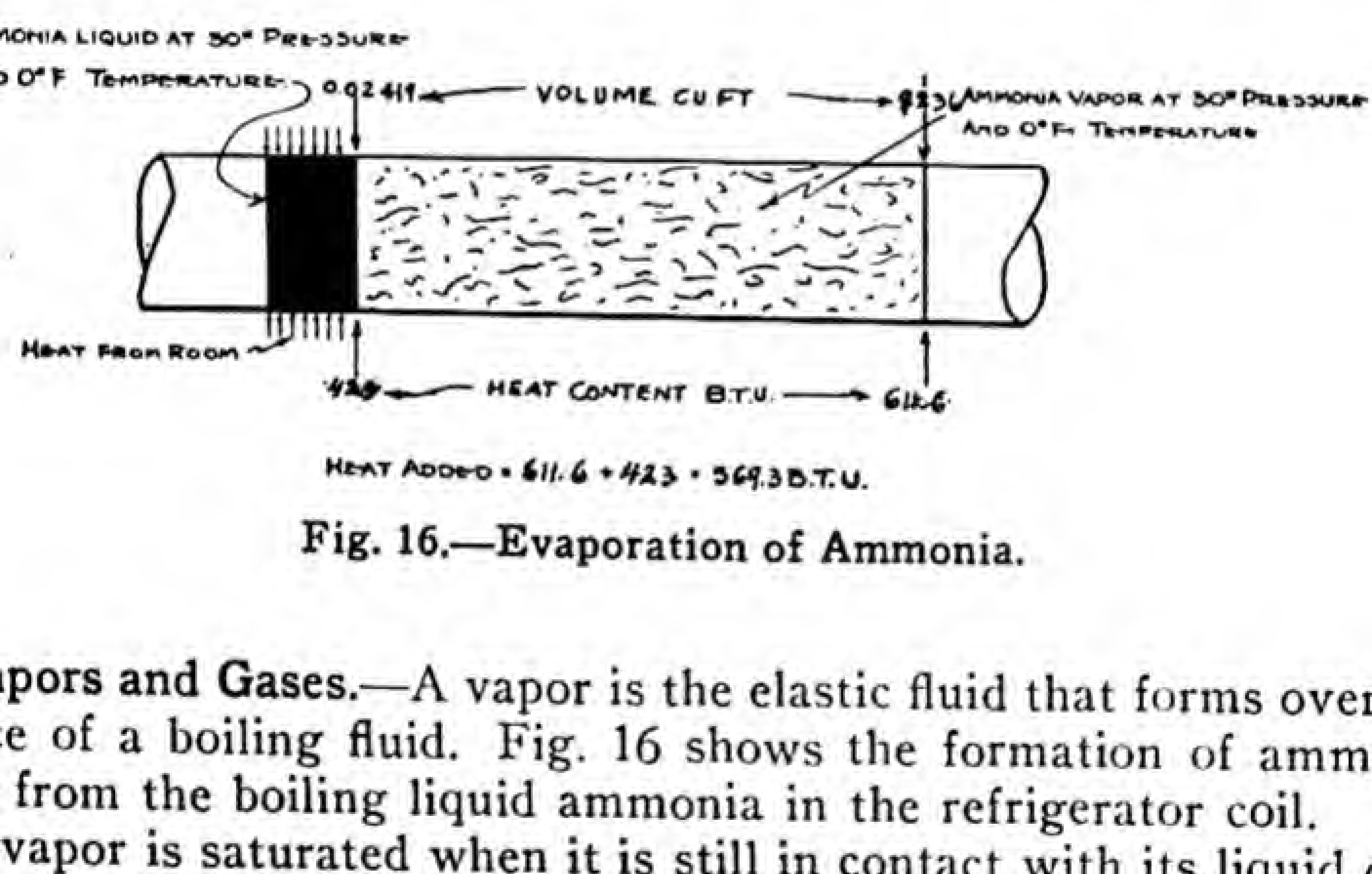


Fig. 16.—Evaporation of Ammonia.

Vapors and Gases.—A vapor is the elastic fluid that forms over the surface of a boiling fluid. Fig. 16 shows the formation of ammonia vapor from the boiling liquid ammonia in the refrigerator coil.

A vapor is saturated when it is still in contact with its liquid or is at the temperature corresponding to that at the surface of the boiling liquid. Vapors which are several degrees above the temperature of the saturation point are termed superheated, while vapors which are many degrees above the temperature of saturation (due to the existing pressure) are called gases.

Refrigeration.—The general conception of the term refrigeration implies the cooling of a body below the temperatures of the surrounding substances by the extraction of heat from the body in question. When it is produced by mechanical means it is termed mechanical refrigeration.

Units of Refrigeration.—The quantity of heat absorbed in a refrigerating process is generally measured in Btu. The commercial unit of refrigeration is the quantity of heat required to melt a ton of pure solid ice into water at 32° F. and is termed the standard commercial ton of refrigeration. One pound of ice will absorb 144 Btu. at 32° F. in melting, which is its latent heat of fusion. Therefore one standard commercial ton of refrigeration is the removal of $2000 \times 144 = 288{,}000$ Btu. The rate of performing refrigeration may be termed refrigeration-power, and is the production of refrigeration at the rate of one standard commercial ton of 288,000 Btu. per day of 24 hours. The refrigeration-power may be found by dividing the total heat transferred in one day by 288,000. Hence, if the apparatus removes 1,000,000 Btu. in 24 hours the refrigeration-power may be found as follows:

$$\text{Refrigeration-power} = 1{,}000{,}000 \div 288{,}000 = 3.47 \text{ tons.}$$

The following table shows the units of refrigeration capacity expressed for different units of time:

1 ton of refrigeration..............................	= 288,000 Btu. per day.
1 ton of refrigeration..............................	= 12,000 Btu. per hour.
1 ton of refrigeration..............................	= 200 Btu. per minute.
1 ton of refrigeration..............................	= 3.3 Btu. per second.

The unit of capacity is made more definite by specifying the temperature or pressure range under which the refrigerating machine is to operate while extracting the heat. Thus the standard rating of a refrigerating machine is the number of standard commercial tons that it actually performs under adopted standard pressures for the refrigerating medium. The inlet pressure of the refrigerant, measured on outside and within 10 ft. of the machine, is that pressure which corresponds to a saturation temperature of 5° F. or −15° C. The outlet pressure is that pressure which corresponds to a saturation temperature of 86° F. or 30° C. The refrigerating machine is defined as the pressure imposing element and is the compressor cylinder of the compression system or the absorber, liquor pump, and generator of the absorption refrigerating system. It should be noted that the above standard rating is applied to only those machines which use a liquefiable vapor.

When the metric system of units is employed, the following equivalents are useful:

1 kilogram-calorie..............................	= 3.968 Btu. = 1 frigorie.
1 ton of refrigeration..............................	= 72,580 frigories per day.
1 ton of refrigeration..............................	= 3,024 frigories per hour.

Ice Making Capacity.—Refrigerating machines and plants are sometimes rated in respect to their ice making capacity. The ice making capacity of a machine is the number of tons of ice which it can

produce in one day of 24 hours, and in general is equal from 50 per cent to 70 per cent of the refrigeration capacity.

The heat required to produce ice is that which is necessary to cool the water to the freezing point, to freeze it, to cool it to temperature of the brine bath, and to cover other losses.

The temperature of the water may be 90° F. while the brine temperature may be 15° F. The heat required to cool water from 90° to 32° F. would be equal to the temperature range multiplied by the specific heat of water. The specific heat of water is 1 Btu. and the temperature range is $90° - 32° = 58°$ F. The heat removed in cooling the water would be equal to $58 \times 1 = 58$ Btu. To freeze 1 lb. of water requires the removal of 144 Btu. The specific heat of ice is approximately 0.5, so that the amount required to cool the ice to the temperature of the brine would be equal to the temperature range multiplied by 0.5. The temperature range is $32° - 15° = 17°$ F., so, the heat required is equal to $17 \times 0.5 = 8.5$ Btu. Total heat required is $58 + 144 + 8.5 = 210.5$ Btu. Ordinarily, approximately 15 to 20 per cent additional heat is allowed to cover such losses as heat transferred through brine tank insulation, heat from cans, meltage, etc. This additional amount would be equal to $210.5 \times 0.15 = 31.5$ Btu. The total amount of refrigeration required would be $210.5 + 31.5 = 242$ Btu. per lb. of ice, or $242 \times 2000 = 484{,}000$ Btu. per ton. Thus 484,000 Btu. are expended to produce a ton of ice that has only a refrigerating effect of $144 \times 2000 = 288{,}000$ Btu.

Also, since a ton of refrigeration is the removal of 288,000 Btu. per day, the ice making capacity would be equal to $288{,}000 \div 484{,}000 = 0.596 = 59.6$ per cent of the refrigeration capacity.

Refrigeration Loads.—It is well to observe the various elements that make up the refrigeration load imposed upon machines operating in cold storage warehouses and similar establishments. In the determination of the refrigeration load the following items are the most important:

1. Refrigeration to cool goods stored.
2. Refrigeration to absorb heat transmitted through cold storage wall and insulation.
3. Refrigeration to offset ventilation losses.
4. Refrigeration to absorb heat generated in the room.

There are many tables, rules, etc., which are intended to indicate the amount of refrigeration required for the above items, but these often fail to give the correct estimate for the particular plant in question. Therefore, it is urgently advised that in every case an estimate of each of the above factors should be made.

The engineer should be well acquainted with all the factors which

impose the refrigeration load upon the equipment. Each factor should be investigated thoroughly so as to produce only the amount of the refrigeration that is necessary to give the desired results.

Cooling of Material.—The heat to be extracted from the materials to be stored depends upon the initial temperature, final temperature in cold storage room, the specific heat and weight and whether or not they are frozen. In cooling the material the temperature is lowered to that of the room and in case the room temperature is a few degrees below the freezing point of the material, the latent heat of fusion must be removed, after which the material is further cooled to the room temperature. To determine the amount of sensible heat to be removed in cooling of the material it is only necessary to multiply together the weight in pounds, the temperature range and the specific heat.

The refrigeration to cool one lb. of beef, to freeze it, and to further cool it to 5° F. may be considered. The initial temperature of the beef is 70° F., the specific heat above the freezing point of approximately 28° F. is 0.77 Btu. per lb. The latent heat of fusion is 102 Btu. per lb. and the specific heat after freezing is 0.41 Btu. per lb. The refrigeration may be tabulated as follows:

Cooling from 70° to 28°	$1 \times (70 - 28) \times 0.77 =$	32.34 Btu.
Freezing	1×102 $=$	102.00 Btu.
Cooling 28° to 5°	$1 \times (28 - 5) \times 0.41 =$	9.43 Btu.
Total cooling per lb.	$=$	143.77

The specific heats before freezing, the latent heats of fusion, and the specific heat after freezing and other data of common commodities may be found in Table 83.

Loss of Refrigeration Through Walls.—Since the temperatures of the cold storage rooms are several degrees below the temperature of the atmosphere, heat will flow by natural tendency from the outside to inside through the insulation and the wall itself. Also, since it is obviously impossible to construct a cold storage room wall that will absolutely stop the flow of heat, the transfer of heat is a continual one. The amount of heat transmitted depends upon the area of the surface in sq. ft., the temperature difference between the outside and the inside, and the thickness and kind of insulation. The rate of heat transmission through the wall depends upon the kind and thickness of insulation, and the rate in Btu. per sq. ft. per degree of temperature difference per hour is termed the coefficient of heat transmission for the building wall. The total amount of heat transmitted through the wall is found by multiplying together the area in sq. ft., the temperature difference between the outside and the inside of the room and heat transfer coefficient.

Thus the heat transmitted through the walls of a room 10 ft. long by 10 ft. wide by 10 ft. high with an outside temperature of 70° F. and an inside temperature of 5° F., with a heat transmission coefficient of the wall equal to 0.10 Btu. per sq. ft. per degree temperature difference per hour would be found as follows:

Area 6 ft. × 10 ft. × 10 ft.	= 600 sq. ft.
Temperature difference (70°—5°)	= 65°
Heat transmission coefficient....................................	= 0.10 Btu.
Total heat transfer, 600 × 65 × 0.10........................	= 3900 Btu.

Ventilation Losses.—The heat to be extracted from the air that is required for ventilation purposes depends upon the size of the room, the temperatures of the atmosphere and the room, and the humidity of the air. The actual cooling amounts to the cooling of the air and the condensation of the moisture that is thrown out as the temperature is lowered. For ordinary cold storage temperatures and outside temperatures it is necessary to allow from 2 to 3 Btu. per cu. ft. of air required for ventilation.

Heat Generated in Rooms.—Refrigeration is required to offset the heat generated in the rooms by workmen, lights, motors, fans, etc. The average workman in a cold storage room will give off to the surrounding air about 500 Btu. per hour. For each watt of capacity in electric lights, 3.41 Btu. should be allowed. The heat developed by motors, fans, etc., will be in proportion to the mechanical equivalent of heat.

QUESTIONS ON CHAPTER II.

1. Define energy, work, and power.
2. What are the units of work and power?
3. State the relations of gauge pressures and absolute pressures.
4. What is heat and what is temperature?
5. How is heat measured?
6. Define specific heat, sensible heat, and latent heat.
7. What is the unit of refrigeration and how is it derived?
8. What is the unit of refrigeration capacity?
9. Explain the relation of the refrigeration and ice making capacities of a refrigerating machine.
10. What are the general factors which make up the refrigeration load on a machine in a cold storage plant?

TABLE 4.—CONVERSION FACTORS.
By The Insulite Company

Multiply	By	To obtain
Acres	43560	Square feet
Acres	4047	Square meters
Acres	4840	Square yards
Ampere-turns	1.257	Gilberts
Ampere-turns per cm	2.540	Ampere-turns per inch
Ampere-turns per inch	0.3937	Ampere-turns per cm
Atmospheres	76.0	Cms. of mercury
Atmospheres	29.92	Inches of mercury
Atmospheres	14.70	Pounds per sq. inch
Atmospheres	1058	Tons per sq. ft.
Board-feet	144 sq. in. × 1 in.	Cubic inches
British thermal units	0.2520	Kilogram-calories
British thermal units	777.5	Foot-pounds
British thermal units	3.927×10^{-4}	Horse-power-hours
British thermal units	1054	Joules
British thermal units	107.5	Kilogram-meters
British thermal units	2.928×10^{-4}	Kilowatt-hours
Btu. per minute	12.96	Foot-pounds per sec.
Btu. per minute	0.02356	Horse-power
Btu. per minute	0.01757	Kilowatts
Btu. per minute	17.57	Watts
Bushels	1.244	Cubic feet
Bushels	2.50	Cubic inches
Bushels	4	Peck
Bushels	16	Pint (dry)
Bushels	32	Quart (dry)
Centimeters	0.3937	Inches
Centimeters	0.01	Meters
Centimeters	10	Millimeters
Centimeters of mercury	0.01316	Atmospheres
Centimeters of mercury	0.4461	Feet of water
Centimeters of mercury	27.85	Pounds per square foot
Centimeters of mercury	0.1934	Pounds per square inch
Centimeters per second	1.969	Feet per minute
Centimeters per second	0.03281	Feet per second
Centimeters per second	0.02237	Miles per hour Centimeter
Centimeters per second	3.72×10^{-4}	Miles per minute
Circular mils	5.067×10^{-6}	Square centimeters
Circular mils	0.7854×10^{-6}	Square inches
Cubic centimeters	3.531×10^{-5}	Cubic feet
Cubic centimeters	6.102×10^{-2}	Cubic inches
Cubic centimeters	10^{-6}	Cubic meters
Cubic centimeters	1.308×10^{-6}	Cubic yards
Cubic centimeters	2.642×10^{-4}	Gallons
Cubic centimeters	10^{-3}	Liters
Cubic centimeters	2.113×10^{-3}	Pints (liq.)
Cubic centimeters	1.057×10^{-3}	Quarts (liq.)
Cubic feet	2.832×10^{4}	Cubic cms
Cubic feet	1728	Cubic inches
Cubic feet	0.02832	Cubic meters
Cubic feet	0.03704	Cubic yards
Cubic feet	7.481	Gallons
Cubic feet	59.84	Pints (liq.)
Cubic feet	29.92	Quarts (liq.)
Cubic feet per min.	472.0	Cubic cms. per sec.
Cubic feet per min.	0.1247	Gallons per sec.

TABLE 4.—CONVERSION FACTORS.—Continued.

Multiply	By	To obtain
Cubic feet per min.	0.4720	Liters per sec.
Cubic feet per min.	62.4	Pounds of water per min.
Cubic inches	16.39	Cubic centimeters
Cubic inches	5.787×10^{-4}	Cubic feet
Cubic inches	1.639×10^{-5}	Cubic meters
Cubic inches	2.143×10^{-5}	Cubic yards
Cubic inches	4.329×10^{-3}	Gallons
Cubic inches	1.639×10^{-2}	Liters
Cubic inches	0.03463	Pints (liq.)
Cubic inches	0.01732	Quarts (liq.)
Cubic meters	35.31	Cubic feet
Cubic meters	61023	Cubic inches
Cubic meters	264.2	Gallons
Cubic meters	2113	Pints (liq.)
Cubic meters	1057	Quarts (liq.)
Degrees (angle)	60	Minutes
Degrees (angle)	0.01745	Radians
Degrees (angle)	3600	Seconds
Degrees per second	0.01745	Radians per second
Degrees per second	0.1667	Revolutions per min.
Degrees per second	0.002778	Revolutions per sec.
Dollars (U.S.)	5.182	Francs (French)
Dollars (U.S.)	4.20	Marks (German)
Dollars (U.S.)	0.2055	Pounds sterling (Brit.)
Dollars (U.S.)	4.11	Shillings (British)
Dynes	1.020×10^{-3}	Grams
Dynes	7.233×10^{-5}	Poundals
Dynes	2.248×10^{-6}	Pounds
Ergs	9.486×10^{-11}	British thermal units
Ergs	1	Dynes-centimeters
Ergs	7.376×10^{-8}	Foot pounds
Ergs	1.020×10^{-3}	Grams-centimeters
Ergs	10^{-7}	Joules
Ergs	2.390×10^{-11}	Kilogram-calories
Ergs per second	5.692×10^{-9}	Btu. per minute
Ergs per second	4.426×10^{-6}	Foot-pounds per min.
Ergs per second	7.37×10^{-8}	Foot-pounds per sec.
Ergs per second	1.341×10^{-10}	Horse-power
Ergs per second	10^{-10}	Kilowatts
Fathoms	6	Feet
Feet	30.48	Centimeters
Feet	0.3048	Meters
Feet of water	0.02950	Atmospheres
Feet of water	0.8826	Inches of mercury
Feet of water	62.43	Pounds per sq. foot
Feet of water	0.4335	Pounds per sq. inch
Feet per min.	0.5080	Centimeters per sec.
Feet per min.	0.01667	Feet per sec.
Feet per min.	0.3048	Meters per minute
Feet per min.	0.01136	Miles per hour
Feet per second	30.48	Centimeters per sec.
Feet per second	0.5921	Knots per hour
Feet per second	18.29	Meters per minute
Feet per second	0.6818	Miles per hour
Foot-pounds	1.28×10^{-3}	British thermal units
Foot-pounds	1.356×10^{7}	Ergs
Foot-pounds	1.356	Joules

TABLE 4.—CONVERSION FACTORS.—Continued.

Multiply	By	To obtain
Foot-pounds	3.241×10^{-4}	Kilogram-calories
Foot-pounds	3.766×10^{-7}	Kilowatts
Foot-pounds per min	1.286×10^{-3}	Btu. per minute
Foot-pounds per min	3.030×10^{-5}	Horse-power
Foot-pounds per min	2.260×10^{-5}	Kilowatts
Foot-pounds per sec	1.818×10^{-3}	Horse-power
Foot-pounds per sec	1.356×10^{-3}	Kilowatts
Gallons	3785	Cubic centimeters
Gallons	0.1337	Cubic feet
Gallons	231	Cubic inches
Gallons	3.785×10^{-3}	Cubic meters
Gallons	3.785	Liters
Gallons per minute	2.228×10^{-3}	Cubic feet per second
Gilberts	0.7958	Ampere-turns
Gilberts per centimeter	2.021	Ampere-turns per inch
Grains (troy)	1	Grains (av.)
Grains (troy)	0.06480	Grams
Grains (troy)	0.04167	Pennyweight (troy)
Grams	980.7	Dynes
Grams	15.43	Grains (troy)
Grams	10^{-3}	Kilograms
Grams	0.03527	Ounces
Grams	0.03215	Ounces (troy)
Grams	2.205×10^{-3}	Pounds
Gram-calories	3.968×10^{-3}	British thermal units
Grams-centimeters	9.302×10^{-8}	British thermal units
Gram-centimeters	980.7	Ergs
Gram-centimeters	7.233×10^{-5}	Foot-pounds
Gram-centimeters	9.807×10^{-5}	Joules
Grams per cm	5.600×10^{-3}	Pounds per inch
Grams per cm	62.43	Pounds per cubic foot
Grams per cm	0.03613	Pounds per cubic inch
Horse-power	42.44	Btu. per minute
Horse-power	2546	Btu. per hour
Horse-power	33000	Foot-pounds per min.
Horse-power	550	Foot-pounds per sec.
Horse-power	0.7457	Kilowatts
Horse-power	745.7	Watts
Horse-power (Boiler)	33520	Btu. per hour
Horse-power-hours	1.98×10^{6}	Foot-pounds
Horse-power-hours	2.684×10^{6}	Joules
Horse-power-hours	0.7457	Kilowatt-hours
Inches	2.540	Centimeters
Inches of mercury	0.03342	Atmospheres
Inches of mercury	1.133	Feet of water
Inches of mercury	70.73	Pounds per square foot
Inches of mercury	0.4912	Pounds per square inch
Inches of water	0.002458	Atmospheres
Inches of water	0.07355	Inches of mercury
Inches of water	0.5781	Ounces per square inch
Inches of water	5.204	Pounds per square foot
Inches of water	0.03613	Pounds per square inch
Joules	9.48×10^{-4}	British thermal units
Joules	10^{-7}	Ergs
Joules	0.7376	Foot-pounds
Joules	2.77×10^{-4}	Watt-hours

TABLE 4.—CONVERSION FACTORS.—(Continued.)

Multiply	By	To obtain
Kilograms	980665	Dynes
Kilograms	2.205	Pounds
Kilogram-calories	3.968	British thermal units
Kilogram-calories	3086	Foot-pounds
Kilogram-meters	9.302×10^{-3}	British thermal units
Kilogram-meters	7.233	Foot-pounds
Kilogram-meters	2.724×10^{-6}	Kilowatt-hours
Kilometers	10^5	Centimeters
Kilometers	3281	Feet
Kilometers	0.6214	Miles
Kilometers per hour	54.68	Feet per minute
Kilometers per hour	0.9113	Feet per second
Kilometers per hour	0.6214	Miles per hour
Kilowatts	56.92	Btu. per minute
Kilowatts	4.42×10^4	Foot-pounds per min.
Kilowatts	1.341	Horse-power
Kilowatt-hours	3415	British thermal units
Kilowatt-hours	2.655×10^6	Foot-pounds
Kilowatt-hours	1.341	Horse-power-hours
Knots	6080	Feet
Knots	1.152	Miles
Knots per hour	1.689	Feet per second
Knots per hour	1.152	Miles per hour
Liters	10^3	Cubic centimeters
Liters	0.03531	Cubic feet
Liters	61.02	Cubic inches
Liters	0.2642	Gallons
Liters	2.113	Pints (liq.)
Liters	1.057	Quarts (liq.)
Liters per min.	4.403×10^{-3}	Gallons per sec.
Meters	3.281	Feet
Meters	39.37	Inches
Meters	1.094	Yards
Meters per min.	3.281	Feet per min.
Meters per min.	0.05468	Feet per sec.
Meters per min.	0.03728	Miles per hour
Meters per sec.	196.8	Feet per min.
Meters per sec.	3.281	Feet per sec.
Meters per sec.	2.237	Miles per hour
Meters per sec.	0.03728	Miles per min.
Miles	1.609×10^5	Centimeters
Miles	5280	Feet
Miles	1.609	Kilometers
Miles	1760	Yards
Miles per hour	88	Feet per min.
Miles per hour	1.467	Feet per sec.
Miles per hour	1.609	Kilometers per hour
Miles per hour	0.8684	Knots per hour
Miner's inch	1.5	Cubic feet per min.
Ounces	437.5	Grains
Ounces	28.35	Grams
Ounces	0.0625	Pounds
Ounces (fluid)	1.805	Cubic inches
Ounces (fluid)	0.02957	Liters
Ounces (fluid)	31.10	Grams
Ounces (troy)	31.10	Grams
Ounces (troy)	0.08333	Pounds (troy)

TABLE 4.—CONVERSION FACTORS.—Concluded.

Multiply	By	To obtain
Ounces per sq. in.	0.0625	Pounds per sq. in.
Pennyweight (troy)	1.555	Grams
Pint (dry)	33.60	Cubic inches
Pint (liq.)	28.87	Cubic inches
Pounds	444823	Dynes
Pounds	7000	Grains
Pounds	453.6	Grams
Pounds	16	Ounces
Pounds (troy)	0.8229	Pounds (av.)
Pounds of water	0.01602	Cubic feet
Pounds of water	27.68	Cubic inches
Pounds of water	0.1198	Gallons
Pounds of water per min.	2.669×10^{-4}	Cubic feet per sec.
Pounds per sq. ft.	0.01602	Feet of water
Pounds per sq. ft.	4.882	Kgs. per sq. meter
Pounds per sq. ft.	6.944×10^{-3}	Pounds per sq. in.
Pounds per sq. in.	0.06804	Atmospheres
Pounds per sq. in.	2.307	Feet of water
Pounds per sq. in.	2.036	Inches of mercury
Quarts (dry)	67.20	Cubic inches
Quarts (liq.)	57.75	Cubic inches
Quires	25	Sheets
Radians	57.30	Degrees
Radians	34.38	Minutes
Radians per sec.	57.30	Degrees per sec.
Radians per sec.	0.1592	Revolutions per sec.
Radians per sec.	9.549	Revolutions per min.
Revolutions	360	Degrees
Revolutions	6.283	Radians
Revolutions per min.	6	Degrees per sec.
Revolutions per min.	0.1047	Radians per sec.
Revolutions per min.	0.01667	Revolutions per sec.
Rods	16.5	Feet
Square centimeters	1.973×10^{5}	Circular mils
Square centimeters	1.076×10^{-3}	Square feet
Square centimeters	0.1550	Square inches
Square feet	2.296×10^{-5}	Acres
Square feet	929.0	Square centimeters
Square feet	144	Square inches
Square feet	0.09290	Square meters
Square feet	3.587×10^{-8}	Square miles
Square inches	1.273×10^{6}	Circular mils
Square inches	6.452	Square centimeters
Square inches	6.944×10^{-3}	Square feet
Square inches	10^{-6}	Square mils
Square kilometers	247.1	Acres
Square kilometers	10.76×10^{6}	Square feet
Square kilometers	0.3861	Square miles
Square meters	2.47×10^{-4}	Acres
Square meters	10.76	Square feet
Square meters	3.86×10^{-7}	Square miles
Square miles	640	Acres
Square miles	27.88×10^{6}	Square feet
Square miles	2.590	Square kilometers
Tons (long)	2240	Pounds
Tons (short)	2000	Pounds

CHAPTER III.

REFRIGERATING MEDIA.

Refrigeration and Refrigerating Media.—Refrigeration has been defined as the cooling of substances below the temperatures of the surroundings by the extraction of heat from the substances. This definition implies that a substance of a lower temperature than the material to be cooled is used to extract the heat from the material in question. The substance that absorbs the heat from the material to be cooled may be termed "refrigerant." The working substances or refrigerants of mechanical refrigeration systems may be considered collectively as refrigerating media. These refrigerating media may be classified into two groups, in respect to the manner of absorption of heat. In the first group are those which cool the material by absorbing their latent heats of evaporation. These refrigerants are liquefiable vapors, such as ammonia, the Freons, carbon dioxide, sulphur dioxide, methyl chloride. In the second group are those which cool by absorbing their sensible heats. These refrigerating media include air, calcium chloride brine, sodium chloride brine.

Possible Refrigerant.—There are many substances which may be used as refrigerant, as will be seen later, ammonia, Freon 12, methyl chloride, carbon dioxide, and sulphur dioxide are now the most widely used in the United States. By using a substance that may be changed from liquid to the vapor state, advantage may be taken of the latent heat of evaporation. The heat that is absorbed in the refrigerator, or the refrigerating effect, may be found by subtracting the heat content of the liquid just before the expansion valve from the heat content of the saturated vapor as it leaves the refrigerator coils. Thus, if the ammonia in the coils has a temperature of 0° F. and the temperature of the liquid just before expansion valve is 85° F. the refrigerating effect would be $611.8 - 137.8 = 474.0$ Btu. per lb. Since the heat content of the vapor leaving the coil is 611.8 and since the heat content of the liquid just before the expansion valve is 137.8, it is evident that the ammonia absorbs the difference or 474.0 Btu. in passing through the refrigerator. On the other hand, if use is made only of the sensible

heat the refrigerating effect is quite small. In this case the actual amount of refrigerating effect may be found by multiplying together the weight of material, the specific heat and the temperature range. A quantity of air in warming from $-50°$ to $0°$ F. may be used to cool a room at $10°$ F. If the specific heat of air is 0.25 Btu. per lb. per deg., the refrigerating effect is

$$1 \times 0.25 \times (-50° - 0°) = 12.5 \text{ Btu. per lb.}$$

In comparison with the refrigerating effect of 474.0 Btu. per lb. for ammonia under the above condition it will be seen that the refrigerating effect of 12.5 Btu. is quite small. To produce an equivalent amount of refrigeration, it is evident that a large volume of air must be circulated.

The possibility of using a given refrigerant for a certain refrigeration problem depends upon many factors. The most important considerations entering into the selection of a suitable refrigerant are the temperature and pressure ranges, the cost of the working fluids, the chemical properties, and the physical properties.

The Temperature and Pressure Range.—In order to cause boiling or evaporation the temperature of the refrigerator must be higher than that of the boiling refrigerant. Also in order to cause condensation or liquefaction the cooling water must be colder than the saturated vapor. Every refrigerant has a range of temperatures corresponding to definite pressures at which it will evaporate and condense. These temperature and pressure relations may be found in the first two columns of the tables on refrigerants. Since water is most commonly used for condensing purposes a refrigerant must be selected that will liquefy with the usual water temperatures and at reasonable pressures. On the other hand, the temperature of evaporation must be obtained at satisfactory pressures. In general, it is more desirable to keep evaporating pressures above that of the atmosphere. This is due to the fact that when the pressures are below atmospheric, air and moisture may be drawn into the system through the stuffing boxes and loose joints. Air and moisture may cause serious operating troubles. The following table shows the boiling temperatures of some of the more common refrigerants at atmospheric pressure, and the absolute pressures of their liquids at 86° F

	Temperatures at Atmospheric Pressure	*Abs. Pressures at 86.0° F. Temp.*
Ammonia	−28.0° F.	169.2 lbs./sq. in.
Carbon Dioxide	−110.0° F.	1039.0 lbs./sq. in.
Sulphur Dioxide	13.8° F.	66.45 lbs./sq. in.
Freon 12	−21.7° F.	107.9 lbs./sq. in.
Methyl Chloride	−10.0° F.	94.7 lbs./sq. in.

In the event that excessively high pressures exist in the condenser, the use of heavy pipes, steel cylinders, special fittings, stuffing boxes, etc., are necessary. It is obvious that these factors must be kept well within reason in commercial apparatus.

Cost of the Working Fluid.—Due to leakage through joints and stuffing boxes, or loss due to accidents or impure condition of refrigerant, it is generally necessary to replenish the charge at certain intervals. It is then apparent that the cost of the working fluid will determine to some extent its commercial application. If the cost of the working fluid was sufficiently low, the liquefiable refrigerants could be wasted to the atmosphere, after they have absorbed their latent heats of evaporation from the materials in the refrigerator.

However, all of the known refrigerants at present cost so much that they must be used over and over again, by repetitions of the working cycle.

The Chemical Properties.—The chemical characteristics of the working fluids are important in the efficient selection of the refrigerant, on account of two considerations. In the first place, the refrigerant must not have a corrosive action upon the various metals which are used in the construction of the system. Although the refrigerants in their pure state do not attack the metals used in the systems generally, such foreign matter as water, oils, etc., in combination with the refrigerant may promote corrosion. In the second place, the fluids must have a strong chemical bond in order to withstand the repeated evaporations, condensations, absorptions, and dissociations of the various cycles of operation. In the compression system the refrigerants may show a tendency to disintegrate into their respective elementary substances. This is due to the fact that the temperatures are fairly high at the end of compression.

Later in this chapter it will be shown that the most important refrigerants have such chemical properties which allow them to be used conveniently in commercial refrigeration systems.

The Physical Properties.—The physical properties of the various available refrigerating media determine their commercial application to a very large extent. The pressure that is required to liquefy the vapors in the condenser affects to a very considerable degree the design of the system. The specific volume per pound of the vapor at the refrigerator pressure, determines to a great extent the amount of piston displacement, and in order for a refrigerant to come into universal use it is necessary to keep this value as low as possible. The magnitude of the latent heat of evaporation and the density per cubic foot affect the quantity of medium to be circulated to produce a given refrigerating effect. The critical temperature of a vapor is the tem-

perature above which it is impossible to condense the vapor by application of pressure. Since the temperature of the saturated vapor in the condenser is several degrees above the temperature of the condenser cooling water, it is evident that the critical temperature must be considered in relation to the maximum temperature of the cooling water.

Relationship of Physical Properties.—In considering the various states of matter it is evident that certain properties of matter determine the state of the material. These important properties of matter are pressure, volume, temperature, heat content and entropy. These may be termed the *cardinal* properties of matter, since their magnitudes determine the characteristics of the material at a given state. Furthermore, when any two of the five cardinal properties are known, the other properties may be found, thus if the pressure and specific volume are known it is possible to calculate the size or magnitude of the temperature, heat content, etc.

The new properties may be determined by calculation or taken from tables of properties of the various substances under consideration. Ammonia vapor at 170 lbs. abs. pressure and at a temperature 86.29° F. may have its temperature increased 153.71° or to 240° F. The following tabulation will show the variation of the values of the cardinal properties:

	State I.	*State II.*
Temperature, deg. F.	86.29	240.
Pressure, lbs. abs.	170.	170.
Specific volume, cu. ft. per lb.	1.764	2.473
Heat content, Btu. per lb.	631.6	730.9
Entropy	1.1900	1.3512

When the pressure of a vapor is increased its temperature, or temperature of its boiling point, is increased proportionally. However, the other properties will be decreased in the same proportion. As an example, ammonia vapor has its pressure increased from 20 lbs. to 170 lbs. abs. pressure. The following tabulation will show the properties at the various states:

	State I.	*State II.*
Pressure, lbs. abs.	20 00	170.00
Temperature, deg. F.	—16.64	86.29
Specific volume, cu. ft. per lb.	13.50.	1.764
Heat content, Btu. per lb.	606.2	631.6
Entropy	1.3700	1.1900

Bureau of Standards Ammonia Tables.—One of the first things that the operating refrigerating engineer should understand is the ammonia tables. While there are a number of refrigerants in use, the greater number of industrial plants use ammonia.

The United States Bureau of Standards, in 1923, published the properties of ammonia, given in Tables 6 to 10, inclusive. Table 6 is a saturated ammonia temperature table, which has even degrees of temperature, ranging from $-60°$ to $125°$ F., along the left- and right-hand sides of the tables. Table 7 is a saturated ammonia absolute pressure table, which shows the properties of ammonia from 5 to 300 lbs. pressure for even pounds of absolute pressure. Table 8 is a saturated ammonia gauge pressure table, which shows the properties of ammonia corresponding to gauge pressures, ranging from 0 to 300 lbs., which shows also the properties of ammonia corresponding to inches of mercury, ranging from 20 to 0 in. of mercury below one standard atmosphere. Table 9 gives the properties of liquid ammonia, ranging from the freezing point at $-107.86°$ F. to the critical temperature at $271.4°$ F.

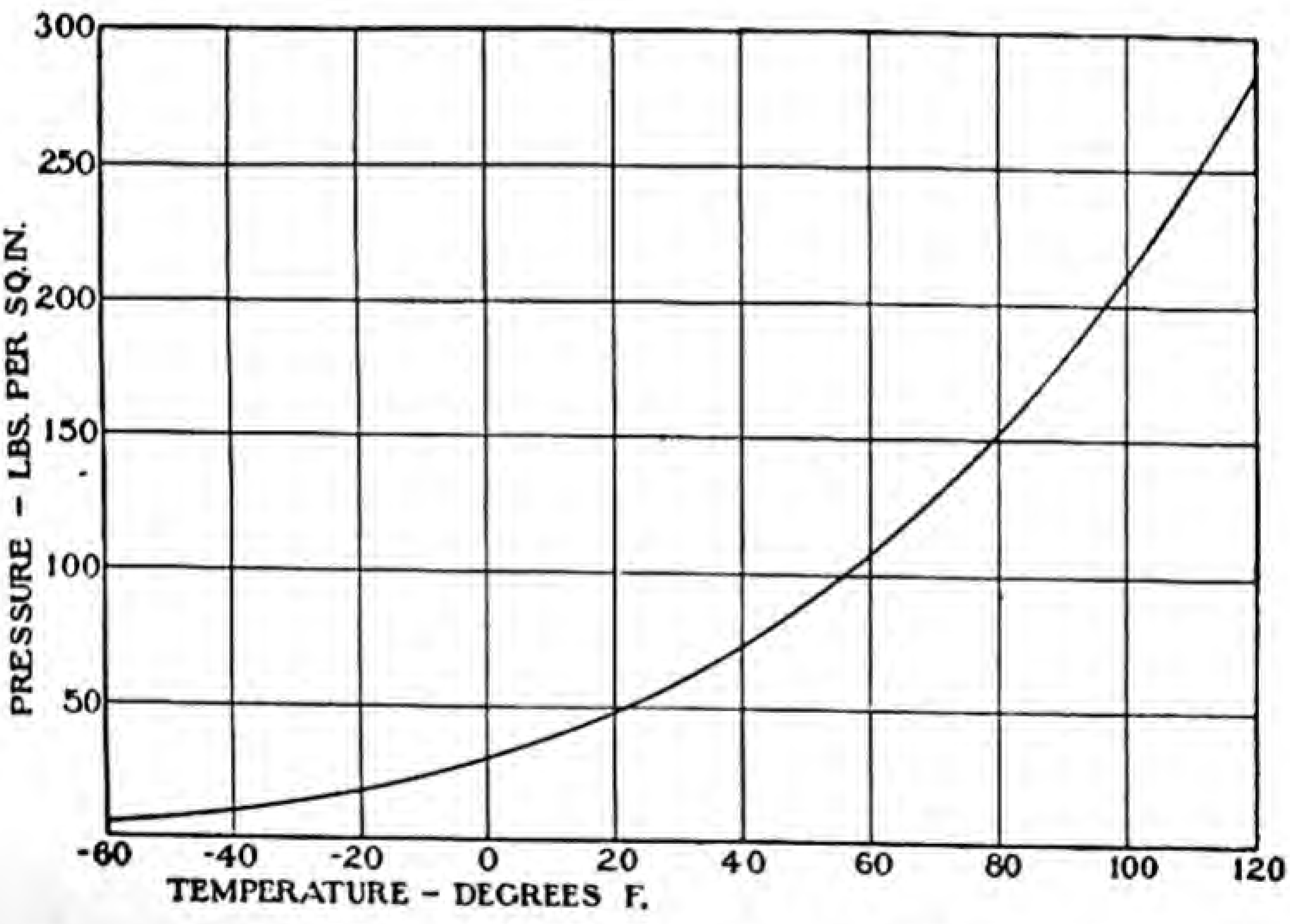


Fig. 17.—Temperature-Pressure Curve for Ammonia.

Table 10 gives the properties of superheated ammonia vapor. Referring to Table 6 again, it is desired to see what the figures in the various columns mean. In Table 6 the first column gives the temperature in degrees Fahrenheit *(t)* of saturated ammonia corresponding to the absolute pressure, lbs. per sq. in. *(p)*, and corresponding to gauge pressure in lbs. per sq. in. *(g.p.)*.

By referring to Table 6, it will be observed that when the temperature of saturated ammonia is increased the corresponding pres-

sure increases proportionately—that is, the higher the temperature the higher the pressure, and vice versa.

By saturated condition is meant the condition at which the ammonia is on the point of evaporating or condensing.

Temperature-Pressure Relationship.—The manner in which the relationship between the temperature and pressure varies is shown graphically by Fig. 17. This figure shows plainly how increasing the temperature of the condensing or boiling point increases the pressure in proportion. The particular relationship existing between the temperature and pressure of ammonia is graphically shown by Fig. 18 for a temperature of saturated ammonia of 70° F.

The liquid ammonia receiver shown in Fig. 18 is partially filled with liquid ammonia. The space immediately above the liquid is filled with the saturated vapor as shown. The thermometer indicates a temperature of 70° F.

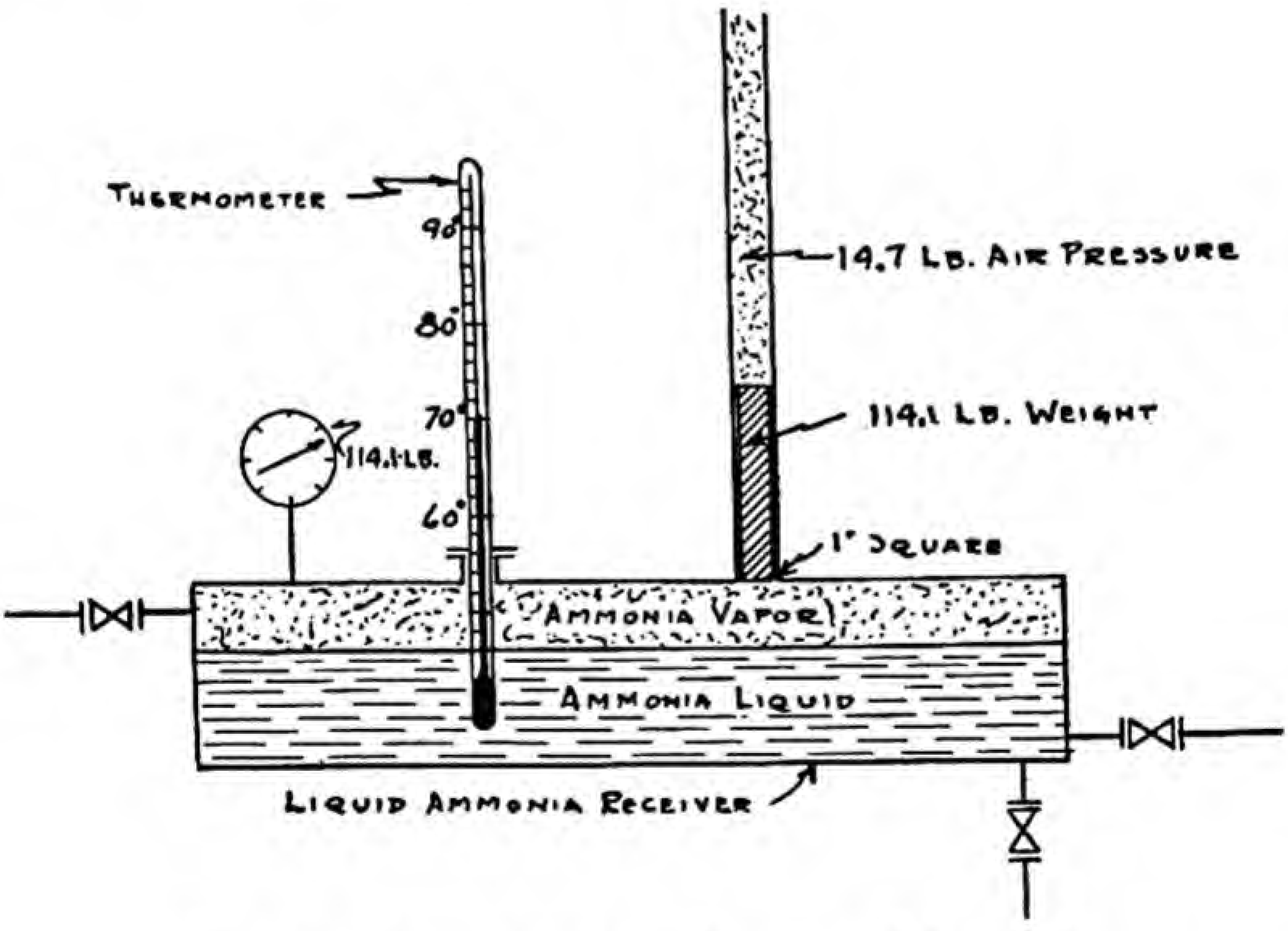


Fig. 18.—Temperature and Pressure of Ammonia.

By referring to the ammonia tables at a point of 70° F. it will be observed that the absolute pressure is given as 128.8, and the gauge pressure is given as 114.1. The meaning of the gauge pressure of 114.1 is explained by referring to Fig. 18. In Fig. 18 a vertical tube, one inch square, is attached to the ammonia cylinder. The top of this tube is open to the atmosphere. In the tube is inserted a weight of

sufficient amount to just balance the pressure within the receiver. When the intensity of pressure of the air upon the weight is 14.7 lbs. per sq. in., then the weight itself would have to contain 114.1 lbs. to balance the pressure within the receiver.

If a pressure gauge is attached to the drum as shown, it will in a similar manner register the pressure of the ammonia vapor above atmospheric pressure, which in this case would be 114.1 lbs.

From Fig. 18 it will be observed also that the total force exerted by the weight in the vertical tube would be equivalent to 114.1 + 14.7 = 128.8 lbs. per sq. in., which is the absolute pressure corresponding to the pressure of the ammonia when maintained at a temperature of 70° F.

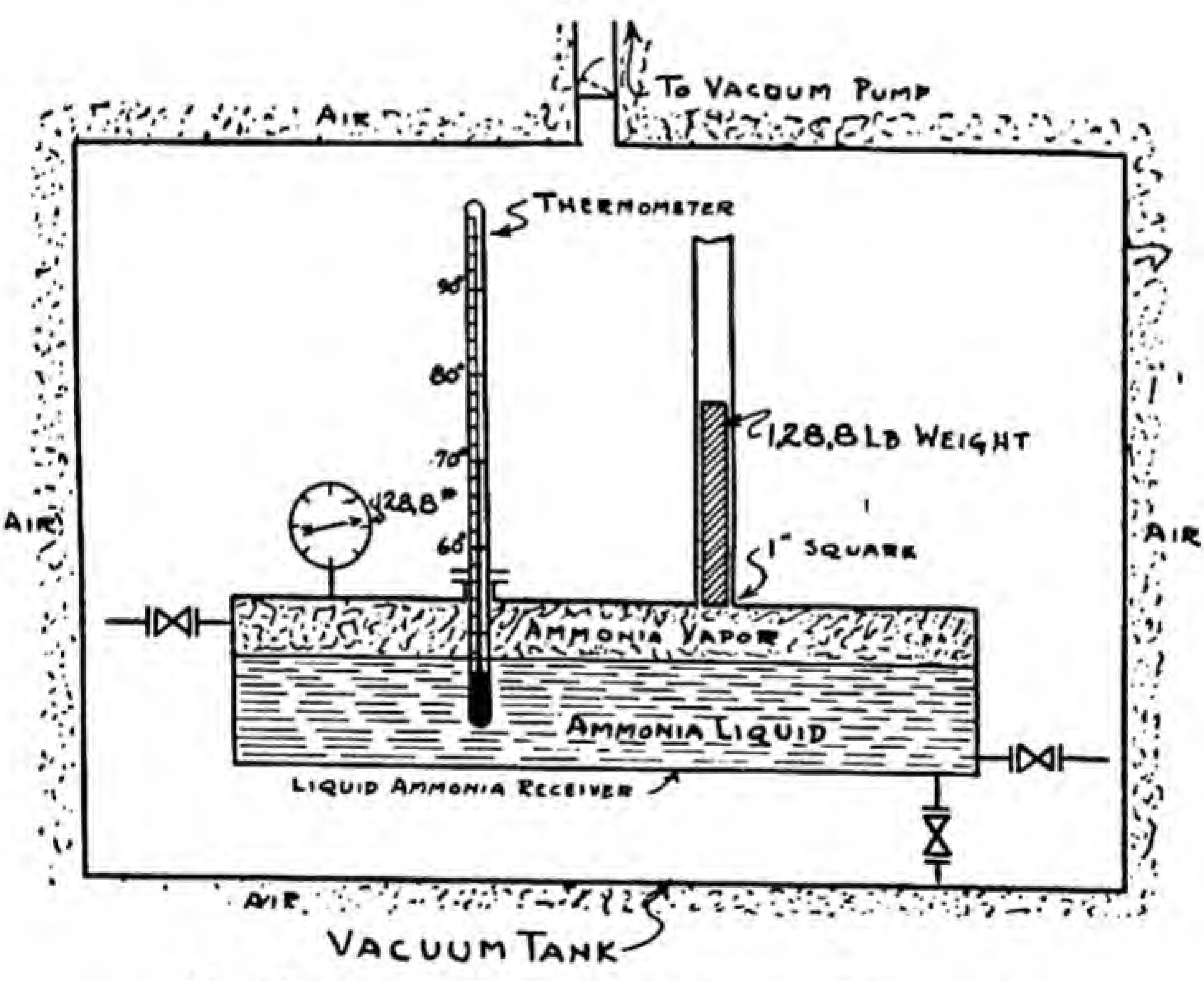


Fig. 19.—Temperature and Pressure of Ammonia.

The relationship between absolute pressure and gauge pressure is more fully illustrated by referring to Fig. 19. In this figure the ammonia receiver and instruments shown in Fig. 18 are put into air-tight vessels, after which a very perfect vacuum is pumped on same. Under this condition the pressure of the air in the vertical tube containing the weight has been removed, so that in order to balance the pressure of the ammonia in the receiver the weight in the tube would have to be increased to 128.8 lbs. This, of course, holds true so long

as the vertical tube has an area of 1 sq. in. A thermometer as indicated would show a temperature of 70° F., and if a pressure gauge was attached to the receiver it would register 128.8 lbs. per sq. in., which corresponds to the intensity of pressure exerted by the weight in the vertical tube upon an area having 1 sq. in. of surface. The point to be borne in mind is that when the temperature of ammonia is 70° F., and when there is both liquid and vapor present—that is, the ammonia is in the saturated condition—then its gauge pressure is always 114.1 lbs. and the corresponding absolute pressure 128.8 lbs., when the atmospheric pressure is 14.7 lbs.

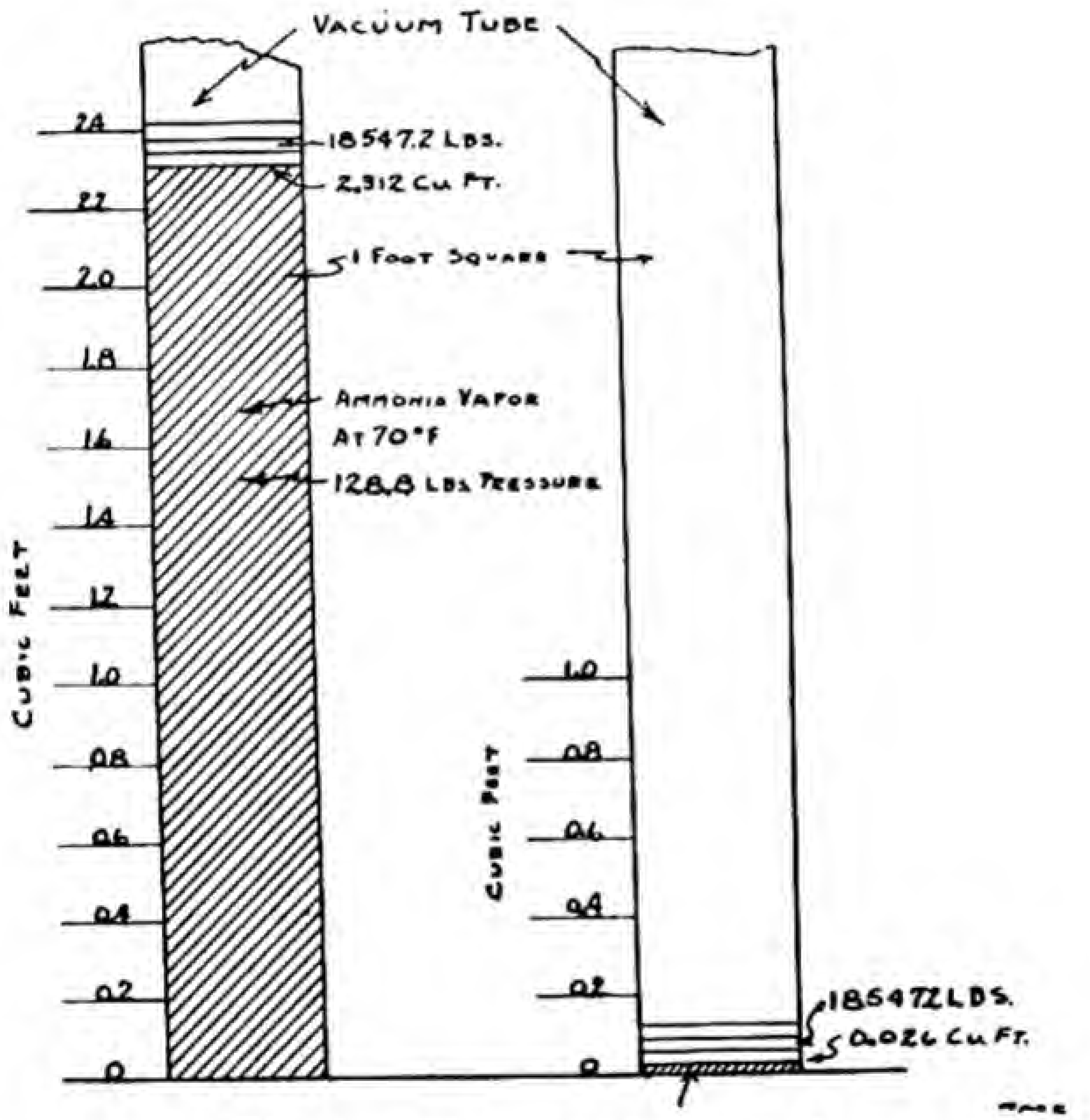


Fig. 20.—Volume of Ammonia.

Specific Volume of Ammonia.—In the third column of Table 6 are given the values of the specific volumes of the vapor *(v)*, expressed in cubic feet per pound. At 70° F. it will be noted that the specific volume of the vapor is 2.312 cu. ft. per lb.

By referring to Table 9, "Properties of Liquid Ammonia" at 70° F., it will be noted that the specific volume of *(v)* of the liquid is 0.0263 cu. ft. per lb. The meaning of the foregoing values is illustrated by Fig. 20.

In Fig. 20 are illustrated two vertical tanks, one foot square, which

contain a vacuum on each side. If a pound of saturated ammonia vapor at a temperature of 70° F. is placed within the tube to the left, as shown, and confined by means of a piston, the height of the piston from the bottom of the tube would be 2.312 ft.

Inasmuch as the pressure inside of the tube originally is a vacuum, and on account of the fact that the saturated ammonia vapor at 70° F. has an intensity of absolute pressure of 128.8 lbs. per sq. in., the weight of the piston to confine the vapor would have to be $144 \times 128.8 = 18547.2$ lbs.

The meaning of the specific volume of a liquid as compared to the specific volume of the vapor under the same conditions is shown by the vacuum tube to the right in Fig. 20.

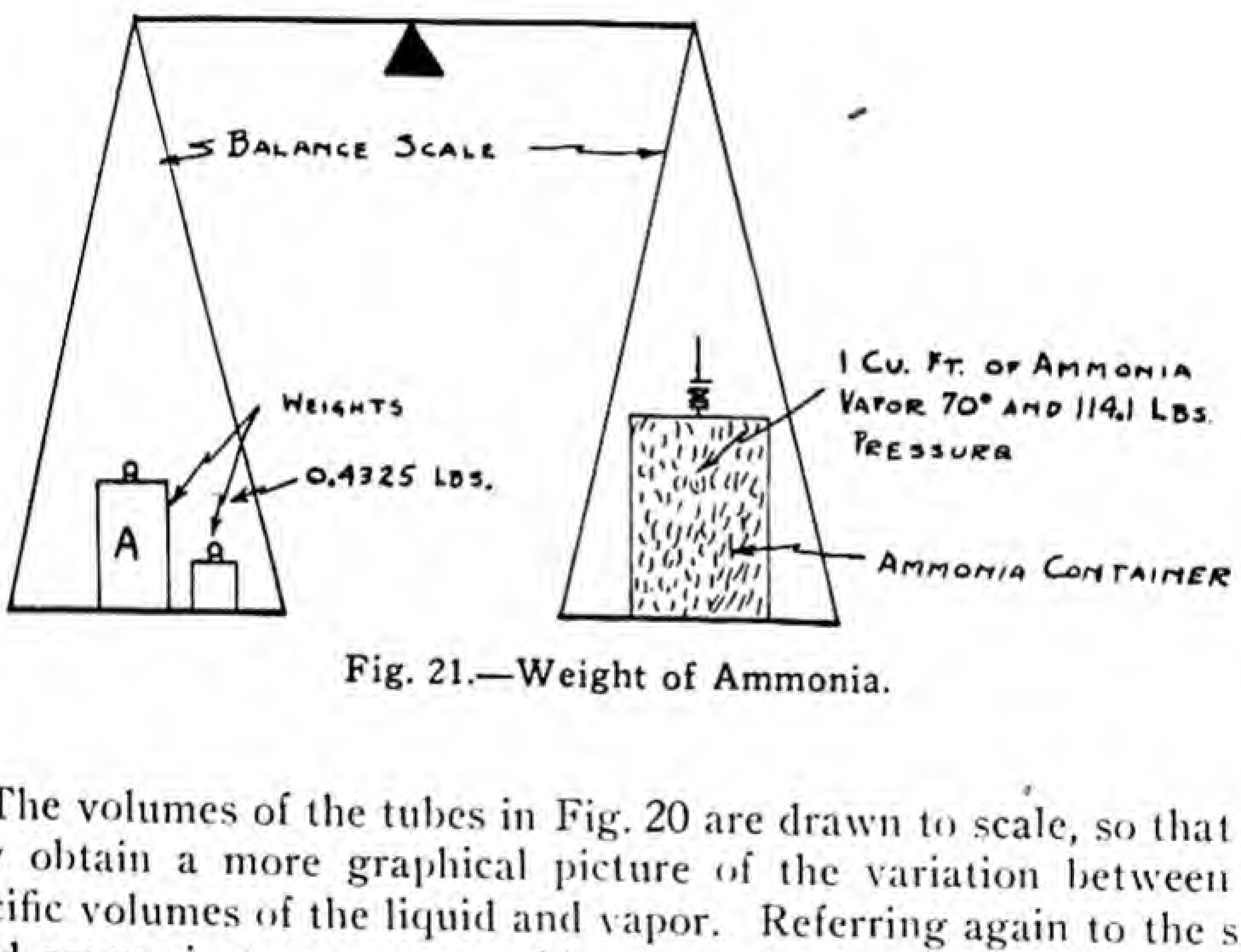


Fig. 21.—Weight of Ammonia.

The volumes of the tubes in Fig. 20 are drawn to scale, so that one may obtain a more graphical picture of the variation between the specific volumes of the liquid and vapor. Referring again to the saturated ammonia temperature table at a temperature of 70° F., in the fifth column, the density of the vapor I/V is given in pounds per cubic foot.

For a temperature of saturated ammonia of 70° F. the density of the vapor is given as 0.4325 lbs. per cu. ft. The meaning of this is illustrated by referring to Fig. 21. In Fig. 21 a vacuum container holding exactly one cubic foot is placed on the platform scales and balanced by means of weight A. The container is then filled with saturated ammonia vapor at a temperature of 70° F. and a gauge pressure of 114.1 lbs. per sq. in. Then, in order to balance these scales, it is necessary to add a weight as shown, equal to 0.4325 lbs. This shows graphically that

the weight of one cubic foot of ammonia vapor at 70° F. and pressure of 114.1 lbs. is 0.4325 lbs. An extra column shown in Table 6, "Saturated Ammonia Temperature Table," gives the heat content of the liquid *(h)* and the heat content of the vapor *(H)* temperature, expressed in Btu. per lb.

Heat Content.—By heat content is meant the heat in a material above a certain reference temperature. For the purpose of the ammonia tables, —40° F. has been selected as a reference point, above which all heat contents are reckoned as being positive, and below which all heat contents are reckoned as negative. Consequently, the heat content of ammonia at any particular temperature means the heat which is required to bring the ammonia to the temperature in question from a temperature of —40° F. It will be remembered, that in the former ammonia tables, as well as in the steam tables, 32° F. was used as a reference temperature. In selecting —40° F. as a reference temperature the Bureau of Standards had in mind the elimination of the addition and subtraction of awkward positive and negative quantities, which is necessary when 32° F. is used as a reference temperature.

Most problems in refrigerating engineering are above —40° F., so that working of negative quantities is practically eliminated. By referring to the sixth column it will be noted that the heat content of the liquid is given as 0 at —40° F., and that for all temperatures above —40° F. the heat contents are positive, and below they are negative. Thus, if the temperature of the liquid ammonia is increased from —40° to 0° F., the heat added is that shown as the heat content of the liquid at 40° F., which in this case is 42.9 Btu. per lb. In like manner, when it is increased from —40 to +50 and +66, the heat contents would be 97.9 and 116 Btu. per lb., respectively. However, at 66° F. the pressure of the ammonia is maintained at 120 lbs. per sq. in. abs., and if heat is still applied to the liquid ammonia it will continue to evaporate at this temperature until the ammonia is entirely evaporated. The heat required to change the state of the liquid ammonia to that of the vapor ammonia is known as the latent heat of evaporation. This is shown by the eighth column of the saturated ammonia temperature table, which gives latent heat *(L)* Btu. per lb.

For a temperature of evaporation of 66° F. it will be noted that the latent heat of evaporation is 512.4 Btu. per lb. This means that 512.4 Btu. will be required to change a pound of liquid ammonia to a pound of vapor ammonia. Then, if it is desired to determine the amount of heat in the ammonia, it is only necessary to add together the heat in the liquid above —40° F. and the latent heat of evaporation. Consequently, the heat content of the saturated ammonia vapor at a temperature of 66° F. would be found as follows: $512.4 + 116 = 628.4$ Btu.,

which corresponds to the heat content of the vapor *(H)* given by the seventh column of the saturated ammonia temperature table. Another way of saying the same thing is to say that the difference between heat content of the saturated vapor and the saturated liquid at the same temperature is the equivalent to the latent heat of evaporation.

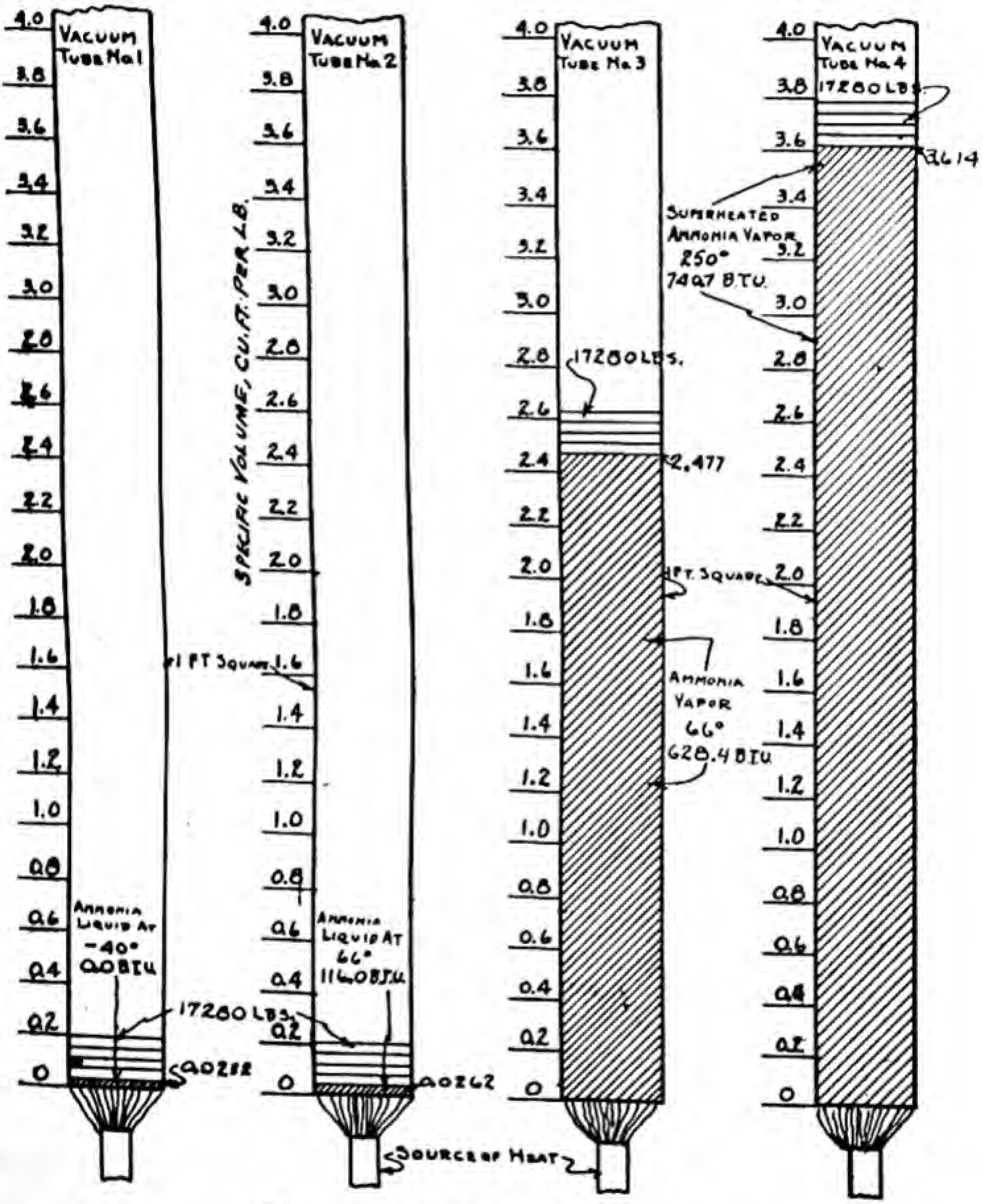


Fig. 22.—Heating of Ammonia.

The particular point to be observed in the foregoing is that the heat content of the saturated ammonia vapor at a temperature of 66° F. and a corresponding pressure of 105.3 lbs. sq. in. gauge is 628.4 Btu. per lb. The saturated ammonia temperature table shows also that for

a temperature of 66° F. that the specific volume of the ammonia vapor is 2.477 cu. ft. per lb. If the saturated ammonia vapor is still maintained at a pressure of 105.3 lbs. per sq. in. gauge, and heat be applied, the temperature will begin to rise immediately. It might be assumed that heat is applied until the temperature rises from 66° to 250° F.

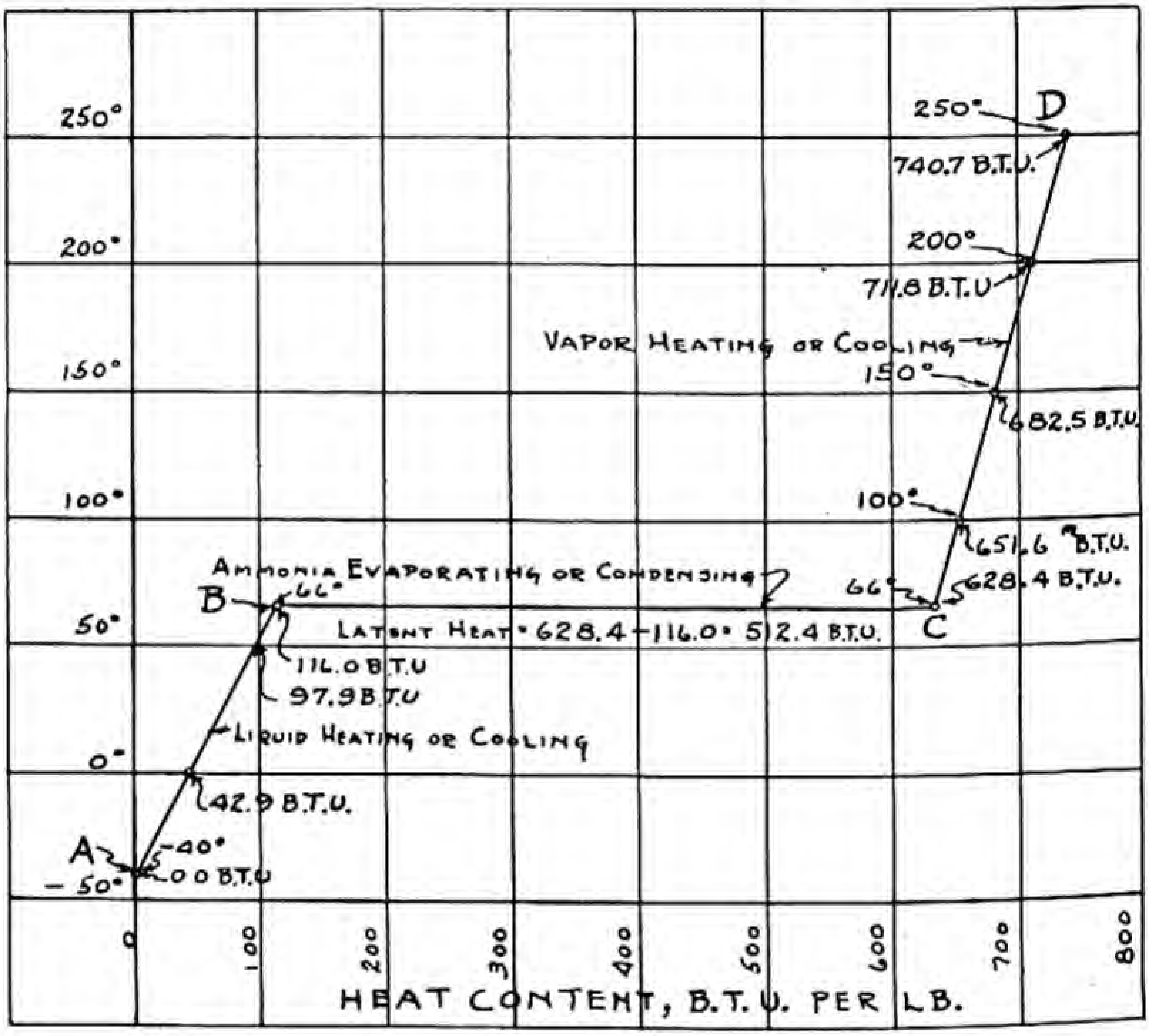


Fig. 23.—Heat Content of Ammonia.

By referring to Table 10, "Properties of Superheated Ammonia Vapor," at a pressure of 120 lbs. per sq. in. abs., and a temperature of 250° F., it will be noted that the heat content of the superheated ammonia vapor is 740.7 Btu. per lb., and the volume has increased to 3.614 cu. ft. per lb.

The heating of ammonia from —40° to 250° F. under the conditions described in the foregoing is illustrated by Fig. 22. Vacuum tube No. 1 contains one pound of liquid ammonia at —40° F., in which its heat content and liquid are shown. The condition of liquid ammonia at 66° F. is shown by vacuum tube No. 2. Here the temperature has increased from —40° up to 66° F., the volume has increased from 0.0232

to 0.0262, and the heat content has increased to 116 Btu. per lb. In vacuum tube No. 3 is shown the condition of the ammonia vapor at 66° F., while the condition of the superheated ammonia vapor at 250° F. is shown as vacuum tube No. 4. The vacuum tubes are one foot square and contain pistons which weigh $144 \times 120 = 17280$ lbs. to confine the ammonia.

The relative height of the pistons in the different tubes illustrates graphically the specific volume of the ammonia at the corresponding conditions. The relationship between the temperature and heat content of ammonia for the foregoing conditions is illustrated graphically by Fig. 23. In this diagram the line *AB* represents the heating of the liquid ammonia from −40° F. to 66° F. The vertical distance in the chart represents temperature, while the horizontal represents heat contents. The line *BC* represents the increase in the heat content during the evaporation period. Line *CD* represents the superheating of the ammonia vapor from 66° to 250° F., at a constant pressure of 120 lbs. per sq. in. Fig. 23 gives a graphical illustration of how the heat of ammonia is increased when its temperature is increased, and a change of state produced also.

Entropy.—The last two columns in Table 6, "Saturated Ammonia Tables" are labeled "Entropy of the Liquid" (s) and "Entropy of the Vapor" (S), expressed in Btu. per lb. per deg. F.; also, in Table 7, the saturated ammonia temperature table, it will be observed that the entropy of the vaporization L/T is given also.

While the name entropy may seem a little difficult to understand, it must be remembered that it is simply a mathematical ratio, which is obtained by dividing the heat added or subtracted during a process by the average absolute temperature during the process. For example, one may refer to the entropy of the liquid ammonia. At −40° F. it will be observed that the entropy is given as 0.0, while at a temperature of 66° F. or a pressure of 120 lbs. it is given as 0.2451.

It will further be noted, also, that the heat content at −40° F. is 0.0, and that the heat content at 66° F. is 116.0 Btu. per lb. The approximate method of determining the entropy for this condition is as follows: The heat added during the process of heating ammonia from −40° F. to 66° F. is, according to ammonia table, 116 Btu. The average temperature during the process would be 13° F. This would correspond to an absolute temperature of $13 + 459.6 = 472.6°$ F. The entropy of the liquid ammonia at a temperature of 60° F. is therefore calculated as follows: $116 \div 472.6 = 0.2452$. This corresponds approximately to the value given for the entropy of liquid at 120 lbs. and 66° F. in Table 7. Also, referring to Table 7, it will be observed that the

latent heat of evaporation is 512.4 Btu. per lb. The absolute temperature during the evaporation would be $66 + 459.61 = 525.6$.

The entropy of the evaporation for this condition is therefore calculated as follows: $512.4 \div 525.6 = 0.9749$, which corresponds with the value given in Table 7, and is known as the entropy of evaporation. The entropy of the saturated vapor must consequently be the sum of the entropy of the liquid and evaporation. The entropy of the vapor then would be calculated as follows: $0.2451 + 0.9749 = 1.2200$.

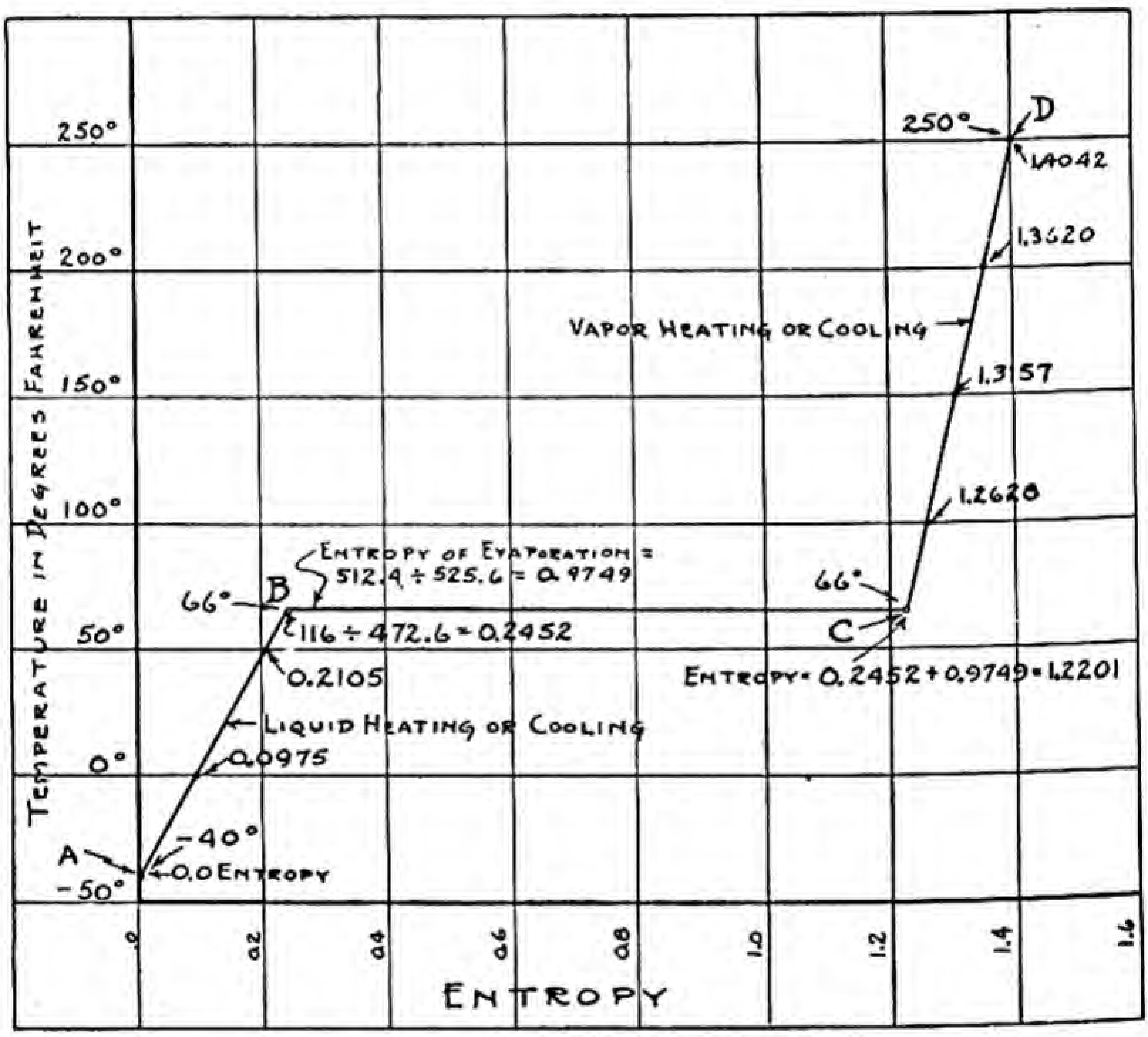


Fig. 24.—Increase of Temperature and Entropy.

In a similar manner, by referring to the superheated ammonia vapor at a pressure of 120 lbs. and a temperature of 250° F., it will be noted that the heat content is 740.7 Btu. per lb. It will be remembered that the heat content of saturated vapor at the same pressure is 628.4. Consequently, the heat added during the superheating process is $740.7 - 628.4 = 112.3$ Btu.

The average temperature during the superheating process would be $66 + 250 \div 2 = 158°$ F. The absolute temperature during the process would be $158 + 459.6 = 617.6°$ F. The entropy of the superheat would

then be as follows: $121.3 \div 617.6 = 0.1820$. The entropy of the superheated vapor would then be found as follows: $1.2200 + 0.1820 = 1.4020$. This corresponds, approximately, to the value given in the superheated table for ammonia vapor at a temperature of 250° F. and pressure of 120 lbs.

A change of entropy of ammonia in accordance with a change of temperature and with the addition of heat, as described in the foregoing, is shown graphically by Fig. 24. This is known as the temperature-entropy chart, vertical distances being plotted according to temperatures, and horizontal distances being plotted according to entropy.

The increase of entropy of the liquid when it is heated from —40° to 66° F. is shown by line *AB*. The entropy of the liquid at 66° F. of 0.2451 is shown by point *B*. The entropy of evaporation is shown by line *BC*. The increase of entropy during the superheating process from 66° to 250° F. is shown by line *CD*. The line *BC* shows how the entropy increases rapidly at constant temperature during the evaporating process. The line *CD* shows how the temperature increases rapidly and the entropy increases slowly during the superheating vapor.

Entropy is just as important a property of ammonia as any other physical characteristic, such as temperature, pressure or volume.

Heat Content Pressure Diagram.—The principal properties of ammonia have been plotted on a diagram which is known as a heat-content pressure-diagram. This is shown in Fig. 25 in the back cover pocket.

In addition to portraying the properties of ammonia in all its phases, the ammonia chart, Fig. 25, prepared by the Bureau of Standards, may be used to show graphically the various cycles of operation which are commonly used in refrigerating systems. Also, the magnitude of the various heat quantities, volumes, pressures, temperatures, may be determined directly from the diagram without reference to the tables of the properties of ammonia. The use of the diagram leads to the determination of fundamental constants of the refrigerating cycles with the minimum amount of calculation. In addition, in its present form, it will yield results which are sufficiently accurate for all practical purposes. From these considerations it is evident that the diagram should be thoroughly understood and used by all who are connected with refrigerating engineering.

Nature of the Chart.—The chart is known as a pressure heat-content diagram. Charts of this type, which are sometimes termed "Mollier Charts," were introduced originally by Mollier. The ordinates or vertical distances in the diagram are pressures which are laid off in proportion to the logarithms of the pressures. The pressures on the ordinates vary from 5 to 300 lbs. abs. The abscissas or horizontal distances of heat contents of ammonia in Btu. per lb. above a reference plane of

$-40°$ F. are laid off on a linear scale, having the dimensions equivalent to 2 and 5 Btu. each. The heat contents shown vary from -26 to 830 Btu. per lb. To bring the diagram within reasonable length, in the center portion of the scale, the heat contents are laid off with divisions that are equivalent to 5 Btu. each. This refers to heat contents shown on the diagram between 200 and 300 Btu. per lb. In a similar manner it should be noted that the pressure scale of this diagram is in reality the logarithms of the pressures having the actual pressures indicated on the scale at the lengths of the various divisions corresponding to the logarithm of the pressures.

Across the left-hand side of the diagram is shown the saturated liquid line, and across the right-hand side of the diagram is shown the saturated vapor line. These two characteristic lines, representing the conditions of saturated liquid and vapor, divide the chart into three parts, and are plotted by laying off heat contents of the liquid and vapor at different pressures from the zero heat content line.

The region in the diagram to the left of the saturated liquid line contains points which represent the conditions of the aftercooled liquid ammonia. Points on the liquid line represent saturation, and points to the left of the liquid represent aftercooled liquid.

Points in the space between the saturated liquid line and the saturated vapor line indicate the conditions of the mixture, of liquid and vapor in the process of evaporation or condensation.

The area of the diagram to the right of the saturated vapor line represents the properties of the superheated vapor, in which points near the saturated vapor line represent small amounts of superheat, while points near the extreme right-hand side of the diagram represent large amounts of super-heat. In addition to the foregoing, characteristic lines, lines of constant quality of the mixture are shown at suitable intervals between the saturated liquid and vapor lines; a family of curves, showing constant volumes, appears in the saturated mixture and superheated vapor regions; across the right-hand side of the saturated mixture region and extending through the superheated vapor region are lines of constant entropy; constant temperature lines pass nearly vertically downward through the liquid region to the saturated liquid curve, from which they pass in a horizontal direction to the saturated vapor curves, and then pass nearly vertically downward through the region of the superheated vapor.

The chemical expression for ammonia is NH_3, where N means nitrogen and H means hydrogen. One atom of nitrogen (N) units with three atoms of hydrogen (H) to form one molecule of ammonia NH_3.

Properties of Ammonia.—It is well to observe the relative characteristics of the various refrigerants. Ammonia is ordinarily harmless

to iron and steel but it will attack copper and copper alloys. Hence, these metals cannot be used in the construction of the system. A solution of ammonia and water such as used in absorption machines may have corroding action on iron and steel, in the presence of foreign matter. The critical pressure and temperature are 1657 lbs. and 271.4° F. Since the critical temperature is high, warm condensing water may be used. But ammonia will begin to disintegrate into nitrogen and hydrogen above 900° F., so that it is advisable to always keep the temperature below 300° F. in order to reduce to a minimum the formation of the permanent gases. The presence of such foreign matter as water, oils, etc., will promote the disintegration action slowly. Ammonia will burn above 900° F., but the combustion does not have the force of an explosion. It is somewhat dangerous to life, due to its corrosive and suffocating action. The compression of the vapor follows the law $PV^{1.28}$ = a constant, where P is the absolute pressure in lbs. per sq. ft. and V is the volume in cu. ft. This means that the product of the pressure, expressed as lbs. per sq. ft. and the volume in cu. ft. raised to the 1.28 power will always equal a constant numerical quantity. Ammonia requires only medium pressures and temperatures in the working cycle.

The following table shows the comparison of the principal properties of saturated ammonia at the temperatures of 5° and 86°:

Temperature, deg. F.	5° F.	86° F.
Pressure, lbs. per sq. in. abs.	34.27	169.2
Pressure, lbs. per sq. in. gauge	19.57	154.5
Specific volume of vapor, cu. ft. per lb.	8.150	1.772
Heat content of liquid, Btu.	48.3	138.9
Heat content of vapor, Btu.	613.3	631.5
Latent heat evaporation, Btu.	565.0	492.6
Entropy of vapor	1.3253	1.1904

Properties of Freon 12.—The most widely used of the Freon group of refrigerants is Freon 12 (CCl_2F_2). It is noncorrosive to steel, iron, brass, copper, aluminum or other metals used in refrigerating systems. It is a medium pressure refrigerant with moderate discharge temperature. At atmospheric pressure the boiling point is −21.6°F. The critical conditions are at 232.7°F. and 582 lbs. pressure abs. Both the liquid and vapor have a relatively high density. The specific gravity of the liquid is 1.44 and of the vapor is 5.2.

Freon 12 has a very slight odor. It is non-toxic and non-combustible. In a high temperature flame it will break down and this fact is used to locate leaks with a halide torch.

The following table shows a comparison of the principal properties of saturated Freon 12 at temperatures of 5° F. and 86° F. Complete data is given in Tables 11 and 12. See also the Pressure-Heat Diagram Fig. 25A and the Heat-Entropy Diagram, Fig. 25B, in the cover pocket.

Temperature, deg. F.	5° F.	86° F.
Pressure, lbs. per sq. in. abs.	26.5	107.9
Pressure, lbs. per sq. in. gauge	11.8	93.2
Specific volume of vapor, cu. ft. per lb.	1.485	0.389
Heat content of liquid, Btu.	9.32	27.77
Heat content of vapor, Btu.	78.79	87.37
Latent heat evaporation, Btu.	69.47	59.65
Entropy of vapor	0.1705	0.1664

Other Freon Refrigerants.—Two other Freon refrigerants are of special interest: Freon 11 which is used in centrifugal compressors and Freon 22 which is especially adapted to low temperature work. Both of them are non-toxic and non-inflammable.

Freon 11 (CCl_3F) is a very low pressure refrigerant with large gas volume at ordinary suction temperatures. Thus it lends itself well to use in centrifugal compressors. The properties of saturated Freon 11, shown in Table 13, at standard conditions are:

Temperature, deg. F.	5° F.	86° F.
Pressure, lbs. per sq. in. abs.	2.93	18.28
Pressure, lbs. per sq. in. gauge	23.95*	3.58
Specific volume of vapor, cu. ft. per lb.	12.27	2.24
Heat content of liquid, Btu. per lb.	8.88	25.34
Heat content of vapor, Btu. per lb.	92.88	102.65
Latent heat of evaporation, Btu. per lb.	84.0	77.31
Entropy of vapor	0.20	0.194

* Inches of mercury below one atmosphere.

Freon 22 ($CHClF_2$) came into wide use during the war for low temperature work. It has a suction pressure higher than ammonia and considerably higher than Freon 12 in the low temperature range although the discharge pressure is not much above ammonia at ordinary condensing temperature. When working at the same temperature levels the discharge temperature is considerably lower than that of ammonia. Table 14 gives the properties of saturated Freon 22 which at 5° and 86° F. are:

Temperature, deg. F.	5° F.	86° F.
Pressure, lbs. per sq. in. abs.	43.12	174.4
Pressure, lbs. per sq. in. gauge	28.42	159.7
Specific volume of vapor, cu. ft. per lb.	1.24	0.311
Heat content of liquid, Btu. per lb.	12.09	36.09
Heat content of vapor, Btu. per lb.	105.29	112.28
Latent heat of evaporation, Btu. per lb.	93.20	76.19
Entropy of vapor	0.228	0.213

Properties of Carbon Dioxide.—The properties of saturated carbon dioxide vapor have been investigated by Mollier, Amagat, Hodsdon, and others. Table 15 shows the properties of this refrigerant for the usual range of working temperatures. Also see the Temperature-Entropy Diagram, Fig. 25C, in the cover pocket. The chemical designation of carbon dioxide is CO_2 where C represents carbon and O represents oxygen.

Carbon dioxide is harmless to iron, steel, copper, etc., and therefore almost any metal may be used in the system. Its critical temperature is 88.4° F. and the corresponding critical pressure is 1071 lbs. abs. Therefore, comparatively cool condensing water must be used. The chemical bond of the molecules is strong and it is naturally non-combustible. Due to the small specific volume, or the large weight per unit volume, the refrigerating effect per unit of displacement is large. This means the compressor cylinder may be made comparatively small for a given tonnage. It, however, requires excessively high pressures, which necessitates the use of steel compressor cylinders, special fittings, packings, etc. The compression of the vapor follows the law of $PV^{1.3}$ = a constant. It has only a suffocating action on life. Carbon dioxide was used only to a small extent in the production of refrigeration in previous years but is being used much more extensively at the present time.

The relative values of the properties at the saturated temperatures of 5° and 86° F. are indicated by the following table:

Temperature, deg. F.	5° F.	86° F.
Pressure, lbs. per sq. in. abs.	331.9	1043.0
Pressure, lbs. per sq. in. gauge	317.2	1028.3
Specific volume of vapor, cu. ft. per lb.	0.2673	0.047
Heat content of liquid Btu. per lb.	21.3	83.3
Latent heat of evaporation Btu.	117.5	27.1
Heat content of vapor Btu.	138.7	110.4
Entropy of vapor (from 32° + 1)	1.2218	1.1381

Properties of Sulphur Dioxide.—The chemical expression for sulphur dioxide is SO_2, where S represents sulphur and O represents oxygen. The properties of saturated sulphur dioxide are shown in Table 16. A pressure-heat diagram, Fig. 26, is in the cover pocket.

The following tabulation will show the variation of the properties at 5° and 86° F.:

Temperature, deg. F.	5° F.	86° F.
Pressure, lbs. per sq. in. abs.	11.81	66.45
Pressure, lbs. per sq. in. gauge	—2.89	51.75
Specific volume, cu. ft. per lb.	6.421	1.185
Heat content of liquid Btu.	14.11	42.12
Latent heat of evaporation Btu.	169.38	142.80
Heat content of vapor Btu.	183.49	184.92
Entropy of vapor	0.3960	0.3495

Sulphur dioxide forms an acid when it is mixed with water, so that all moisture must be excluded from the system. Iron, steel and copper may be used in the construction of the system. Its critical condition exists at 311° F. and 1160 lbs. pressure. Therefore, warm condenser water may be used. The chemical bond is strong and it will, therefore, withstand repeated evaporations and compressions easily. It is, of course, non-combustible. The refrigerating effect per unit volume is

low, so that the compressor cylinders must be comparatively large. The pressures of the condensing vapors are quite low, being from 40 to 60 lbs. gauge. The compression of the vapor follows the law of $PV^{1.26}$ = a constant. It is somewhat dangerous to life due to its corrosive and suffocating action. This refrigerant is used principally in small machines, such as the domestic type.

Properties of Methyl Chloride.—The various thermal properties of methyl chloride have been investigated by Gouault, Holst, Herter, and others. The chemical formula for methyl chloride is CH_3Cl. The properties of the fluid are shown by Table 17. The variations of the properties of methyl chloride at 5° and 86° F. are shown by the following tabulation:

Temperature, deg. F.	5° F.	86° F.
Pressure, lbs. per sq. in. abs.	21.15	94.7
Pressure, lbs. per sq. in. gauge	6.45	80.0
Specific volume of vapor, cu. ft. per lb.	4.47	1.08
Heat content of liquid	16.2	46.67
Latent heat of evaporation	180.7	159.13
Heat content of vapor	196.9	205.8
Entropy of vapor	0.425	0.389

This refrigerant is neutral to iron, steel, brass and copper but it should never be used in systems containing aluminum, zinc or magnesium. Due to the medium pressure ranges, the discharge temperature is generally low, but, if the temperature increases, methyl chloride has a tendency to decompose into carbon, hydrogen and chlorine. Chlorine, of course, will attack the metals of the system. The critical condition exists at 286.9° F. and 1073 lbs. pressure abs. It is noncombustible at ordinary temperatures, but will burn when ignited with a flame. The condensing pressure will vary 60 to 90 lbs. gauge. The compression of the vapor follows the law of $PV^{1.20}$ = a constant.

Methyl chloride is toxic and has a very faint odor. Where more than a few pounds of it are used good ventilation should be provided. The field of application of methyl chloride is comparatively narrow, being used only in the smaller plants, principally in commercial work.

Comparative Value of Refrigerants.—The following tabulation shows the comparison of the five principal refrigerating media, in respect to the pounds of refrigerant per minute required to be vaporized to produce one ton of refrigeration, the theoretical displacement of the compressor per ton of refrigeration and theoretical horsepower per ton of refrigeration. The values are for an evaporator temperature of 5° F. and a temperature of 86° F. in the saturated portion of the condenser.

The pounds of refrigerant per minute per ton varies from 0.422 lbs. of ammonia to 3.9 lbs. of Freon 12. Sulphur dioxide and methyl chloride require about the same as shown by line No. 4 of the table. The

theoretical compressor displacement per min. per ton varies from 0.942 cu. ft. for carbon dioxide to 9.05 for sulphur dioxide. The actual displacement will be 15 to 25 per cent more than these theoretical values. The low displacement for carbon dioxide is due to its low specific volume, while the large displacement for sulphur dioxide is due to its large specific volume. The theoretical horsepower per ton of refrigeration is

TABLE 5. CHARACTERISTICS OF REFRIGERANTS.

	NH_3	CO_2	SO_2	CCl_2F_2	CH_3Cl
1. Heat content of vapor, leaving evaporator, Btu.	613.3	102.1	183.49	78.8	196.9
2. Heat content of liquid, leaving condenser, Btu.	138.9	45.45	42.12	27.8	46.67
3. Refrigerating effect, Btu.	474.4	56.65	141.37	51.0	150.2
4. Pounds of fluid per min. per ton refrigeration	0.422	3.53	1.41	3.9	1.33
5. Specific volume of vapor in refrigerator	8.15	0.2673	6.42	1.48	4.47
6. Displacement per min. per ton cu. ft.	3.43	0.942	9.05	5.8	6.0
7. Mean effective pressure, lbs. per sq. in.	66.0	435.0	24.4	40.3	36.8
8. Theoretical horsepower per ton of refrigeration	0.989	1.78	0.964	0.996	0.965

shown by line No. 8. It will be noted that the horsepower per ton of refrigeration is nearly the same for most of the fluids, while that for carbon dioxide is quite high. Generally speaking, the horsepower per ton of refrigeration should be constant under a given set of conditions, regardless of the fluid used. The exact reason for carbon dioxide requiring so much power is not clearly understood, but no doubt must be in some part due to the high pressures of carbon dioxide.

Aqua Ammonia.—Absorption refrigerating machines using a solution of ammonia and water are of considerable importance. This method of producing refrigeration is used in many plants at present. In order to thoroughly understand the principles of operation of the machine it is necessary to have a good working knowledge of the properties of solutions of ammonia and water. The properties of aqua ammonia have been investigated by Starr, Mollier, Macintire and others. A solution of 30 per cent by weight ammonia (and 70 per cent by weight water) is termed 30 per cent concentration. The temperature at which a solution of the liquor of a given concentration will boil depends upon the pressure. Increasing the pressures raises the boiling point in the same proportion. The temperatures at which the solution will boil depend upon the concentration when the pressure is constant. Increasing the strength lowers the boiling point proportionally.

Tables 18 and 19 give the temperature-pressure-concentration properties of aqua ammonia for ordinary ranges of values. The figures in the body of the tables give the boiling temperatures corresponding to the concentrations and absolute pressures.

Properties of Brines.—In some types of refrigerating plants, it is desirable to place the evaporating coils in a brine solution, and then pump the cold brine through coils located in the rooms or space to be cooled. The brine solutions are made by dissolving some common salt, such as sodium chloride or calcium chloride in water. The addition of a solid to a liquid lowers its freezing point. The amount of salt to be added to the water depends upon the freezing temperature of the brine that is desired. The freezing temperature of the brines should be at least 5° to 10° F. below the lowest ammonia temperature used. The physical properties of calcium chloride brine are shown by Table 20, and the properties of common salt brine, or sodium chloride brine are shown by Table 21.

An inspection of the freezing temperature of these brines will show that calcium chloride brine should be used for low temperature work, and that either calcium or sodium chloride brines may be used for high temperature refrigerating work. The cost of calcium chloride is higher than that of common salt but this disadvantage is partially offset by the fact that approximately 25 per cent less weight of calcium chloride is required.

QUESTIONS ON CHAPTER III.

1. What is the function of a refrigerant?
2. What is the advantage of using a liquefiable fluid?
3. Explain how the working pressures affect the selection of a suitable refrigerant.
4. How do the various temperatures affect the selection?
5. State briefly the relation of the principal properties of vapors and gases.
6. State some of the reasons for the large use of ammonia as a refrigerant.
7. Why are such refrigerants as Freon 12, sulphur dioxide, and methyl chloride used in the smaller machines such as the domestic type?
8. Discuss the use of brines for transmitting refrigeration.
9. In what applications of refrigeration is air used to transmit refrigeration?
10. State the relation of the principal properties of a solution of ammonia and water.

TABLE 6.—SATURATED AMMONIA: TEMPERATURE TABLE.

Temp. °F.	Pressure.		Volume vapor. ft³/lb.	Density vapor. lbs./ft.³	Heat content.		Latent heat. Btu./lb.	Entropy.		Temp. °F.
	Absolute. lbs./in.²	Gage. lbs./in.²			Liquid. Btu./lb.	Vapor. Btu./lb.		Liquid. Btu./lb.°F.	Vapor. Btu./lb.°F.	
t	p	$g.p.$	V	$1/V$	h	H	L	s	S	t
−60	5.55	*18.6	44.73	0.02235	−21.2	589.6	610.8	−0.0517	1.4769	**−60**
−59	5.74	*18.2	43.37	.02306	−20.1	590.0	610.1	−.0490	.4741	−59
−58	5.93	*17.8	42.05	.02378	−19.1	590.4	609.5	−.0464	.4713	−58
−57	6.13	*17.4	40.79	.02452	−18.0	590.8	608.8	−.0438	.4686	−57
−56	6.33	*17.0	39.56	.02528	−17.0	591.2	608.2	−.0412	.4658	−56
−55	6.54	*16.6	38.38	0.02605	−15.9	591.6	607.5	−0.0386	1.4631	**−55**
−54	6.75	*16.2	37.24	.02685	−14.8	592.1	606.9	.0360	.4604	−54
−53	6.97	*15.7	36.15	.02766	−13.8	592.4	606.2	−.0334	.4577	−53
−52	7.20	*15.3	35.09	.02850	−12.7	592.9	605.6	−.0307	.4551	−52
−51	7.43	*14.8	34.06	.02936	−11.7	593.2	604.9	−.0281	.4524	−51
−50	7.67	*14.3	33.08	0.03023	−10.6	593.7	604.3	−0.0256	1.4497	**−50**
−49	7.91	*13.8	32.12	.03113	−9.6	594.0	603.6	−.0230	.4471	−49
−48	8.16	*13.3	31.20	.03205	−8.5	594.4	602.9	−.0204	.4445	−48
−47	8.42	*12.8	30.31	.03299	−7.4	594.9	602.3	−.0179	.4419	−47
−46	8.68	*12.2	29.45	.03395	−6.4	595.2	601.6	−.0153	.4393	−46
−45	8.95	*11.7	28.62	0.03494	−5.3	595.6	600.9	−0.0127	1.4368	**−45**
−44	9.23	*11.1	27.82	.03595	−4.3	596.0	600.3	−.0102	.4342	−44
−43	9.51	*10.6	27.04	.03698	−3.2	596.4	599.6	−.0076	.4317	−43
−42	9.81	*10.0	26.29	.03804	−2.1	596.8	598.9	−.0051	.4292	−42
−41	10.10	*9.3	25.56	.03912	−1.1	597.2	598.3	−.0025	.4267	−41
−40	10.41	*8.7	24.86	0.04022	0.0	597.6	597.6	0.0000	1.4242	**−40**
−39	10.72	*8.1	24.18	.04135	1.1	598.0	596.9	.0025	.4217	−39
−38	11.04	*7.4	23.53	.04251	2.1	598.3	596.2	.0051	.4193	−38
−37	11.37	*6.8	22.89	.04369	3.2	598.7	595.5	.0076	.4169	−37
−36	11.71	*6.1	22.27	.04489	4.3	599.1	594.8	.0101	.4144	−36
−35	12.05	*5.4	21.68	0.04613	5.3	599.5	594.2	0.0126	1.4120	**−35**
−34	12.41	*4.7	21.10	.04739	6.4	599.9	593.5	.0151	.4096	−34
−33	12.77	*3.9	20.54	.04868	7.4	600.2	592.8	.0176	.4072	−33
−32	13.14	*3.2	20.00	.04999	8.5	600.6	592.1	.0201	.4048	−32
−31	13.52	*2.4	19.48	.05134	9.6	601.0	591.4	.0226	.4025	−31
−30	13.90	*1.6	18.97	0.05271	10.7	601.4	590.7	0.0250	1.4001	**−30**
−29	14.30	*0.8	18.48	.05411	11.7	601.7	590.0	.0275	.3978	−29
−28	14.71	0.0	18.00	.05555	12.8	602.1	589.3	.0300	.3955	−28
−27	15.12	0.4	17.54	.05701	13.9	602.5	588.6	.0325	.3932	−27
−26	15.55	0.8	17.09	.05850	14.9	602.8	587.9	.0350	.3909	−26
−25	15.98	1.3	16.66	0.06003	16.0	603.2	587.2	0.0374	1.3886	**−25**
−24	16.42	1.7	16.24	.06158	17.1	603.6	586.5	.0399	.3863	−24
−23	16.88	2.2	15.83	.06317	18.1	603.9	585.8	.0423	.3840	−23
−22	17.34	2.6	15.43	.06479	19.2	604.3	585.1	.0448	.3818	−22
−21	17.81	3.1	15.05	.06644	20.3	604.6	584.3	.0472	.3796	−21
−20	18.30	3.6	14.68	0.06813	21.4	605.0	583.6	0.0497	1.3774	**−20**
−19	18.79	4.1	14.32	.06985	22.4	605.3	582.9	.0521	.3752	−19
−18	19.30	4.6	13.97	.07161	23.5	605.7	582.2	.0545	.3729	−18
−17	19.81	5.1	13.62	.07340	24.6	606.1	581.5	.0570	.3708	−17
−16	20.34	5.6	13.29	.07522	25.6	606.4	580.8	.0594	.3686	−16
−15	20.88	6.2	12.97	0.07709	26.7	606.7	580.0	0.0618	1.3664	**−15**
−14	21.43	6.7	12.66	.07898	27.8	607.1	579.3	.0642	.3643	−14
−13	21.99	7.8	12.36	.08092	28.9	607.5	578.6	.0666	.3621	−13
−12	22.56	7.9	12.06	.08289	30.0	607.8	577.8	.0690	.3600	−12
−11	23.15	8.5	11.78	.08490	31.0	608.1	577.1	.0714	.3579	−11
−10	23.74	9.0	11.50	0.08695	32.1	608.5	576.4	0.0738	1.3558	**−10**

* Inches of mercury below one standard atmosphere (29.92 in.).

TABLE 6.—Continued—SATURATED AMMONIA: TEMPERATURE TABLE.

Temp. °F.	Pressure.		Volume vapor. ft³/lb.	Density vapor. lbs./ft.³	Heat content.		Latent heat. Btu./lb.	Entropy.		Temp. °F.
	Absolute. lbs./in.²	Gage. lbs./in.²			Liquid. Btu./lb.	Vapor. Btu./lb.		Liquid. Btu./lb.°F.	Vapor. Btu./lb.°F.	
t	p	$g.\ p.$	V	$1/V$	h	H	L	s	S	t
−10	23.74	9.0	11.50	0.08695	32.1	608.5	576.4	0.0738	1.3558	**−10**
−9	24.35	9.7	11.23	.08904	33.2	608.8	575.6	.0762	.3537	−9
−8	24.97	10.3	10.97	.09117	34.3	609.2	574.9	.0786	.3516	−8
−7	25.61	10.9	10.71	.09334	35.4	609.5	574.1	.0809	.3495	−7
−6	26.26	11.6	10.47	.09555	36.4	609.8	573.4	.0833	.3474	−6
−5	26.92	12.2	10.23	0.09780	37.5	610.1	572.6	0.0857	1.3454	**−5**
−4	27.59	12.9	9.991	.1001	38.6	610.5	571.9	.0880	.3433	−4
−3	28.28	13.6	9.763	.1024	39.7	610.8	571.1	.0904	.3413	−3
−2	28.98	14.3	9.541	.1048	40.7	611.1	570.4	.0928	.3393	−2
−1	29.69	15.0	9.326	.1072	41.8	611.4	569.6	.0951	.3372	−1
0	30.42	15.7	9.116	0.1097	42.9	611.8	568.9	0.0975	1.3352	**0**
1	31.16	16.5	8.912	.1122	44.0	612.1	568.1	.0998	.3332	1
2	31.92	17.2	8.714	.1148	45.1	612.4	567.3	.1022	.3312	2
3	32.69	18.0	8.521	.1174	46.2	612.7	566.5	.1045	.3292	3
4	33.47	18.8	8.333	.1200	47.2	613.0	565.8	.1069	.3273	4
5	34.27	19.6	8.150	0.1227	48.3	613.3	565.0	0.1092	1.3253	**5**
6	35.09	20.4	7.971	.1254	49.4	613.6	564.2	.1115	.3234	6
7	35.92	21.2	7.798	.1282	50.5	613.9	563.4	.1138	.3214	7
8	36.77	22.1	7.629	.1311	51.6	614.3	562.7	.1162	.3195	8
9	37.63	22.9	7.464	.1340	52.7	614.6	561.9	.1185	.3176	9
10	38.51	23.8	7.304	0.1369	53.8	614.9	561.1	0.1208	1.3157	**10**
11	39.40	24.7	7.148	.1399	54.9	615.2	560.3	.1231	.3137	11
12	40.31	25.6	6.996	.1429	56.0	615.5	559.5	.1254	.3118	12
13	41.24	26.5	6.847	.1460	57.1	615.8	558.7	.1277	.3099	13
14	42.18	27.5	6.703	.1492	58.2	616.1	557.9	.1300	.3081	14
15	43.14	28.4	6.562	0.1524	59.2	616.3	557.1	0.1323	1.3062	**15**
16	44.12	29.4	6.425	.1556	60.3	616.6	556.3	.1346	.3043	16
17	45.12	30.4	6.291	.1590	61.4	616.9	555.5	.1369	.3025	17
18	46.13	31.4	6.161	.1623	62.5	617.2	554.7	.1392	.3006	18
19	47.16	32.5	6.034	.1657	63.6	617.5	553.9	.1415	.2988	19
20	48.21	33.5	5.910	0.1692	64.7	617.8	553.1	0.1437	1.2969	**20**
21	49.28	34.6	5.789	.1728	65.8	618.0	552.2	.1460	.2951	21
22	50.36	35.7	5.671	.1763	66.9	618.3	551.4	.1483	.2933	22
23	51.47	36.8	5.556	.1800	68.0	618.6	550.6	.1505	.2915	23
24	52.59	37.9	5.443	.1837	69.1	618.9	549.8	.1528	.2897	24
25	53.73	39.0	5.334	0.1875	70.2	619.1	548.9	0.1551	1.2879	**25**
26	54.90	40.2	5.227	.1913	71.3	619.4	548.1	.1573	.2861	26
27	56.08	41.4	5.123	.1952	72.4	619.7	547.3	.1596	.2843	27
28	57.28	42.6	5.021	.1992	73.5	619.9	546.4	.1618	.2825	28
29	58.50	43.8	4.922	.2032	74.6	620.2	545.6	.1641	.2808	29
30	59.74	45.0	4.825	0.2073	75.7	620.5	544.8	0.1663	1.2790	**30**
31	61.00	46.3	4.730	.2114	76.8	620.7	543.9	.1686	.2773	31
32	62.29	47.6	4.637	.2156	77.9	621.0	543.1	.1708	.2755	32
33	63.59	48.9	4.547	.2199	79.0	621.2	542.2	.1730	.2738	33
34	64.91	50.2	4.459	.2243	80.1	621.5	541.4	.1753	.2721	34
35	66.26	51.6	4.373	0.2287	81.2	621.7	540.5	0.1775	1.2704	**35**
36	67.63	52.9	4.289	.2332	82.3	622.0	539.7	.1797	.2686	36
37	69.02	54.3	4.207	.2377	83.4	622.2	538.8	.1819	.2669	37
38	70.43	55.7	4.126	.2423	84.6	622.5	537.9	.1841	.2652	38
39	71.87	57.2	4.048	.2470	85.7	622.7	537.0	.1863	.2635	39
40	73.32	58.6	3.971	0.2518	86.8	623.0	536.2	0.1885	1.2618	**40**

TABLE 6.—Continued—SATURATED AMMONIA: TEMPERATURE TABLE.

Temp. °F.	Pressure.		Volume vapor. ft.³/lb.	Density vapor. lbs./ft.³	Heat content.		Latent heat. Btu./lb.	Entropy.		Temp. °F.
	Absolute. lbs./in.²	Gage. lbs./in.²			Liquid. Btu./lb.	Vapor. Btu./lb.		Liquid. Btu./lb.°F.	Vapor. Btu./lb.°F.	
t	p	$g.p.$	V	$1/V$	h	H	L	s	S	t
40	73.32	58.6	3.971	0.2518	86.8	623.0	536.2	0.1885	1.2618	**40**
41	74.80	60.1	3.897	.2566	87.9	623.2	535.3	.1908	.2602	41
42	76.31	61.6	3.823	.2616	89.0	623.4	534.4	.1930	.2585	42
43	77.83	63.1	3.752	.2665	90.1	623.7	533.6	.1952	.2568	43
44	79.38	64.7	3.682	.2716	91.2	623.9	532.7	.1974	.2552	44
45	80.96	66.3	3.614	0.2767	92.3	624.1	531.8	0.1996	1.2535	**45**
46	82.55	67.9	3.547	.2819	93.5	624.4	530.9	.2018	.2519	46
47	84.18	69.5	3.481	.2872	94.6	624.6	530.0	.2040	.2502	47
48	85.82	71.1	3.418	.2926	95.7	624.8	529.1	.2062	.2486	48
49	87.49	72.8	3.355	.2981	96.8	625.0	528.2	.2083	.2469	49
50	89.19	74.5	3.294	0.3036	97.9	625.2	527.3	0.2105	1.2453	**50**
51	90.91	76.2	3.234	.3092	99.1	625.5	526.4	.2127	.2437	51
52	92.66	78.0	3.176	.3149	100.2	625.7	525.5	.2149	.2421	52
53	94.43	79.7	3.119	.3207	101.3	625.9	524.6	.2171	.2405	53
54	96.23	81.5	3.063	.3265	102.4	626.1	523.7	.2192	.2389	54
55	98.06	83.4	3.008	0.3325	103.5	626.3	522.8	0.2214	1.2373	**55**
56	99.91	85.2	2.954	.3385	104.7	626.5	521.8	.2236	.2357	56
57	101.8	87.1	2.902	.3446	105.8	626.7	520.9	.2257	.2341	57
58	103.7	89.0	2.851	.3508	106.9	626.9	520.0	.2279	.2325	58
59	105.6	90.9	2.800	.3571	108.1	627.1	519.0	.2301	.2310	59
60	107.6	92.9	2.751	0.3635	109.2	627.3	518.1	0.2322	1.2294	**60**
61	109.6	94.9	2.703	.3700	110.3	627.5	517.2	.2344	.2278	61
62	111.6	96.9	2.656	.3765	111.5	627.7	516.2	.2365	.2262	62
63	113.6	98.9	2.610	.3832	112.6	627.9	515.3	.2387	.2247	63
64	115.7	101.0	2.565	.3899	113.7	628.0	514.3	.2408	.2231	64
65	117.8	103.1	2.520	0.3968	114.8	628.2	513.4	0.2430	1.2216	**65**
66	120.0	105.3	2.477	.4037	116.0	628.4	512.4	.2451	.2201	66
67	122.1	107.4	2.435	.4108	117.1	628.6	511.5	.2473	.2186	67
68	124.3	109.6	2.393	.4179	118.3	628.8	510.5	.2494	.2170	68
69	126.5	111.8	2.352	.4251	119.4	628.9	509.5	.2515	.2155	69
70	128.8	114.1	2.312	0.4325	120.5	629.1	508.6	0.2537	1.2140	**70**
71	131.1	116.4	2.273	.4399	121.7	629.3	507.6	.2558	.2125	71
72	133.4	118.7	2.235	.4474	122.8	629.4	506.6	.2579	.2110	72
73	135.7	121.0	2.197	.4551	124.0	629.6	505.6	.2601	.2095	73
74	138.1	123.4	2.161	.4628	125.1	629.8	504.7	.2622	.2080	74
75	140.5	125.8	2.125	0.4707	126.2	629.9	503.7	0.2643	1.2065	**75**
76	143.0	128.3	2.089	.4786	127.4	630.1	502.7	.2664	.2050	76
77	145.4	130.7	2.055	.4867	128.5	630.2	501.7	.2685	.2035	77
78	147.9	133.2	2.021	.4949	129.7	630.4	500.7	.2706	.2020	78
79	150.5	135.8	1.988	.5031	130.8	630.5	499.7	.2728	.2006	79
80	153.0	138.3	1.955	0.5115	132.0	630.7	498.7	0.2749	1.1991	**80**
81	155.6	140.9	1.923	.5200	133.1	630.8	497.7	.2769	.1973	81
82	158.3	143.6	1.892	.5287	134.3	631.0	496.7	.2791	.1962	82
83	161.0	146.3	1.861	.5374	135.4	631.1	495.7	.2812	.1947	83
84	163.7	149.0	1.831	.5462	136.6	631.3	494.7	.2833	.1933	84
85	166.4	151.7	1.801	0.5552	137.8	631.4	493.6	0.2854	1.1918	**85**

TABLE 6.—Continued—SATURATED AMMONIA: TEMPERATURE TABLE.

Temp. °F.	Pressure.		Volume vapor. ft.³/lb.	Density vapor. lbs./ft.³	Heat content.		Latent heat. Btu./lb.	Entropy.		Temp. °F.
	Absolute. lbs./in.²	Gage. lbs./in.²			Liquid. Btu./lb.	Vapor. Btu./lb.		Liquid. Btu./lb.°F.	Vapor. Btu./lb.°F.	
t	*p*	*g.p.*	*V*	*1/V*	*h*	*H*	*L*	*s*	*S*	*t*
85	166.4	151.7	1.801	0.5552	137.8	631.4	493.6	0.2854	1.1918	**85**
86	169.2	154.5	1.772	.5643	138.9	631.5	492.6	.2875	.1904	86
87	172.0	157.3	1.744	.5735	140.1	631.7	491.6	.2895	.1889	87
88	174.8	160.1	1.716	.5828	141.2	631.8	490.6	.2917	.1875	88
89	177.7	163.0	1.688	.5923	142.4	631.9	489.5	.2937	.1860	89
90	180.6	165.9	1.661	0.6019	143.5	632.0	488.5	0.2958	1.1846	**90**
91	183.6	168.9	1.635	.6116	144.7	632.1	487.4	.2979	.1832	91
92	186.6	171.9	1.609	.6214	145.8	632.2	486.4	.3000	.1818	92
93	189.6	174.9	1.584	.6314	147.0	632.3	485.3	.3021	.1804	93
94	192.7	178.0	1.559	.6415	148.2	632.5	484.3	.3041	.1789	94
95	195.8	181.1	1.534	0.6517	149.4	632.6	483.2	0.3062	1.1775	**95**
96	198.9	184.2	1.510	.6620	150.5	632.6	482.1	.3083	.1761	96
97	202.1	187.4	1.487	.6725	151.7	632.8	481.1	.3104	.1747	97
98	205.3	190.6	1.464	.6832	152.9	632.9	480.0	.3125	.1733	98
99	208.6	193.9	1.441	.6939	154.0	632.9	478.9	.3145	.1719	99
100	211.9	197.2	1.419	0.7048	155.2	633.0	477.8	0.3166	1.1705	**100**
101	215.2	200.5	1.397	.7159	156.4	633.1	476.7	.3187	.1691	101
102	218.6	203.9	1.375	.7270	157.6	633.2	475.6	.3207	.1677	102
103	222.0	207.3	1.354	.7384	158.7	633.3	474.6	.3228	.1663	103
104	225.4	210.7	1.334	.7498	159.9	633.4	473.5	.3248	.1649	104
105	228.9	214.2	1.313	0.7615	161.1	633.4	472.3	0.3269	1.1635	**105**
106	232.5	217.8	1.293	.7732	162.3	633.5	471.2	.3289	.1621	106
107	236.0	221.3	1.274	.7852	163.5	633.6	470.1	.3310	.1607	107
108	239.7	225.0	1.254	.7972	164.6	633.6	469.0	.3330	.1593	108
109	243.3	228.6	1.235	.8095	165.8	633.7	467.9	.3351	.1580	109
110	247.0	232.3	1.217	0.8219	167.0	633.7	466.7	0.3372	1.1566	**110**
111	250.8	236.1	1.198	.8344	168.2	633.8	465.6	.3392	.1552	111
112	254.5	239.8	1.180	.8471	169.4	633.8	464.4	.3413	.1538	112
113	258.4	243.7	1.163	.8600	170.6	633.9	463.3	.3433	.1524	113
114	262.2	247.5	1.145	.8730	171.8	633.9	462.1	.3453	.1510	114
115	266.2	251.5	1.128	0.8862	173.0	633.9	460.9	0.3474	1.1497	**115**
116	270.1	255.4	1.112	.8996	174.2	634.0	459.8	.3495	.1483	116
117	274.1	259.4	1.095	.9132	175.4	634.0	458.6	.3515	.1469	117
118	278.2	263.5	1.079	.9269	176.6	634.0	457.4	.3535	.1455	118
119	282.3	267.6	1.063	.9408	177.8	634.0	456.2	.3556	.1441	119
120	286.4	271.7	1.047	0.9549	179.0	634.0	455.0	0.3576	1.1427	**120**
121	290.6	275.9	1.032	.9692	180.2	634.0	453.8	.3597	.1414	121
122	294.8	280.1	1.017	.9837	181.4	634.0	452.6	.3618	.1400	122
123	299.1	284.4	1.002	.9983	182.6	634.0	451.4	.3638	.1386	123
124	303.4	288.7	0.987	1.0132	183.9	634.0	450.1	.3659	.1372	124
125	307.8	293.1	0.973	1.028	185.1	634.0	448.9	0.3679	1.1358	**125**

TABLE 7.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SATURATED AMMONIA: ABSOLUTE PRESSURE TABLE.

Pressure (abs.), lbs./in.²	Temp. °F.	Volume vapor. ft.³/lb.	Density vapor. lbs./ft.³	Heat content. Liquid. Btu./lb.	Heat content. Vapor. Btu./lb.	Latent heat. Btu./lb.	Entropy Liquid. Btu./lb. °F.	Entropy Evap. Btu./lb. °F.	Entropy Vapor. Btu./lb. °F.	Pressure (abs.), lbs./in.²
p	t	V	$1/V$	h	H	L	s	L/T	S	p
5.0	−63.11	49.31	0.02029	−24.5	588.3	612.8	−0.0599	1.5456	1.4857	**5.0**
5.5	−60.27	45.11	.02217	−21.5	589.5	611.0	− .0524	.5301	.4777	5.5
6.0	−57.64	41.59	.02405	−18.7	590.6	609.3	− .0455	.5158	.4703	6.0
6.5	−55.18	38.59	.02591	−16.1	591.6	607.7	− .0390	.5026	.4636	6.5
7.0	−52.88	36.01	.02777	−13.7	592.5	606.2	− .0330	.4904	.4574	7.0
7.5	−50.70	33.77	0.02962	−11.3	593.4	604.7	−0.0274	1.4790	1.4516	**7.5**
8.0	−48.64	31.79	.03146	− 9.2	594.2	603.4	− .0221	.4683	.4462	8.0
8.5	−46.69	30.04	.03329	− 7.1	595.0	602.1	− .0171	.4582	.4411	8.5
9.0	−44.83	28.48	.03511	− 5.1	595.7	600.8	− .0123	.4486	.4363	9.0
9.5	−43.05	27.08	.03693	− 3.2	596.4	599.6	− .0077	.4396	.4319	9.5
10.0	−41.34	25.81	0.03874	− 1.4	597.1	598.5	−0.0034	1.4310	1.4276	**10.0**
10.5	−39.71	24.66	.04055	+ 0.3	597.7	597.4	+ .0007	.4228	.4235	10.5
11.0	−38.14	23.61	.04235	2.0	598.3	596.3	.0047	.4149	.4196	11.0
11.5	−36.62	22.65	.04414	3.6	598.9	595.3	.0085	.4074	.4159	11.5
12.0	−35.16	21.77	.04593	5.1	599.4	594.3	.0122	.4002	4124	12.0
12.5	−33.74	20.96	0.04772	6.7	600.0	593.3	0.0157	1.3933	1.4090	**12.5**
13.0	−32.37	20.20	.04950	8.1	600.5	592.4	.0191	.3866	.4057	13.0
13.5	−31.05	19.50	.05128	9.6	601.0	591.4	.0225	.3801	.4026	13.5
14.0	−29.76	18.85	.05305	10.9	601.4	590.5	.0257	.3739	.3996	14.0
14.5	−28.51	18.24	.05482	12.2	601.9	589.7	.0288	.3679	.3967	14.5
15.0	−27.29	17.67	0.05658	13.6	602.4	588.8	0.0318	1.3620	1.3938	**15.0**
15.5	−26.11	17.14	.05834	14.8	602.8	588.0	.0347	.3564	.3911	15.5
16.0	−24.95	16.64	.06010	16.0	603.2	587.2	.0375	.3510	.3885	16.0
16.5	−23.83	16.17	.06186	17.2	603.6	586.4	.0403	.3456	.3859	16.5
17.0	−22.73	15.72	.06361	18.4	604.0	585.6	.0430	.3405	.3835	17.0
17.5	−21.66	15.30	0.06535	19.6	604.4	584.8	0.0456	1.3354	1.3810	**17.5**
18.0	−20.61	14.90	.06710	20.7	604.8	584.1	.0482	.3305	.3787	18.0
18.5	−19.59	14.53	.06884	21.8	605.1	583.3	.0507	.3258	.3765	18.5
19.0	−18.58	14.17	.07058	22.9	605.5	582.6	.0531	.3211	.3742	19.0
19.5	−17.60	13.83	.07232	23.9	605.8	581.9	.0555	.3166	.3721	19.5
20.0	−16.64	13.50	0.07405	25.0	606.2	581.2	0.0578	1.3122	1.3700	**20.0**
20.5	−15.70	13.20	.07578	26.0	606.5	580.5	.0601	.3078	.3679	20.5
21.0	−14.78	12.90	.07751	27.0	606.8	579.8	.0623	.3036	.3659	21.0
21.5	−13.87	12.62	.07924	27.9	607.1	579.2	.0645	.2995	.3640	21.5
22.0	−12.98	12.35	.08096	28.9	607.4	578.5	.0666	.2955	.3621	22.0
22.5	−12.11	12.09	0.08268	29.8	607.7	577.9	0.0687	1.2915	1.3602	**22.5**
23.0	−11.25	11.85	.08440	30.8	608.1	577.3	.0708	.2876	.3584	23.0
23.5	−10.41	11.61	.08612	31.7	608.3	576.6	.0728	.2838	.3566	23.5
24.0	− 9.58	11.39	.08783	32.6	608.6	576.0	.0748	.2801	.3549	24.0
24.5	− 8.76	11.17	.08955	33.5	608.9	575.4	.0768	.2764	.3532	24.5
25.0	− 7.96	10.96	0.09126	34.3	609.1	574.8	0.0787	1.2728	1.3515	**25.0**
25.5	− 7.17	10.76	.09297	35.2	609.4	574.2	.0805	.2693	.3498	25.5
26.0	− 6.39	10.56	.09468	36.0	609.7	573.7	.0824	.2658	.3482	26.0
26.5	− 5.63	10.38	.09638	36.8	609.9	573.1	.0842	.2625	.3467	26.5
27.0	− 4.87	10.20	.09809	37.7	610.2	572.5	.0860	.2591	.3451	27.0
27.5	− 4.13	10.02	0.09979	38.4	610.4	572.0	0.0878	1.2558	1.3436	**27.5**
28.0	− 3.40	9.853	.1015	39.3	610.7	571.4	.0895	.2526	.3421	28.0
28.5	− 2.68	9.691	.1032	40.0	610.9	570.9	.0912	.2494	.3406	28.5
29.0	− 1.97	9.534	.1049	40.8	611.1	570.3	.0929	.2463	.3392	29.0
29.5	− 1.27	9.383	.1066	41.6	611.4	569.8	.0945	.2433	.3378	29.5
30.0	− 0.57	9.236	0.1083	42.3	611.6	569.3	0.0962	1.2402	1.3361	**30.0**

TABLE 7.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SATURATED AMMONIA: ABSOLUTE PRESSURE TABLE.—(Continued.)

Pressure (abs.), lbs./in.²	Temp. °F.	Volume vapor. ft.³/lb.	Density vapor. lbs./ft.³	Heat content. Liquid. Btu./lb.	Heat content. Vapor. Btu./lb.	Latent heat. Btu./lb.	Entropy. Liquid. Btu./lb. °F.	Entropy. Evap. Btu./lb. °F.	Entropy. Vapor. Btu./lb. °F.	Pressure (abs.), lbs./in.²
p	t	V	$1/V$	h	H	L	s	L/T	S	p
30	−0.57	9.236	0.1083	42.3	611.6	569.3	0.0962	1.2402	1.3364	30
31	+0.79	8.955	.1117	43.8	612.0	568.2	.0993	.2343	.3336	31
32	2.11	8.693	.1150	45.2	612.4	567.2	.1024	.2286	.3310	32
33	3.40	8.445	.1184	46.6	612.8	566.2	.1055	.2230	.3285	33
34	4.66	8.211	.1218	48.0	613.2	565.2	.1084	.2176	.3260	34
35	5.89	7.991	0.1251	49.3	613.6	564.3	0.1113	1.2123	1.3236	35
36	7.09	7.782	.1285	50.6	614.0	563.4	.1141	.2072	.3213	36
37	8.27	7.584	.1319	51.9	614.3	562.4	.1168	.2022	.3190	37
38	9.42	7.396	.1352	53.2	614.7	561.5	.1195	.1973	.3168	38
39	10.55	7.217	.1386	54.4	615.0	560.6	.1221	.1925	.3146	39
40	11.66	7.047	0.1419	55.6	615.4	559.8	0.1246	1.1879	1.3125	40
41	12.74	6.885	.1452	56.8	615.7	558.9	.1271	.1833	.3104	41
42	13.81	6.731	.1486	57.9	616.0	558.1	.1296	.1788	.3084	42
43	14.85	6.583	.1519	59.1	616.3	557.2	.1320	.1745	.3065	43
44	15.88	6.442	.1552	60.2	616.6	556.4	.1343	.1703	.3046	44
45	16.88	6.307	0.1586	61.3	616.9	555.6	0.1366	1.1661	1.3027	45
46	17.87	6.177	.1619	62.4	617.2	554.8	.1389	.1620	.3009	46
47	18.84	6.053	.1652	63.4	617.4	554.0	.1411	.1580	.2991	47
48	19.80	5.934	.1685	64.5	617.7	553.2	.1433	.1540	.2973	48
49	20.74	5.820	.1718	65.5	618.0	552.5	.1454	.1502	.2956	49
50	21.67	5.710	0.1751	66.5	618.2	551.7	0.1475	1.1464	1.2939	50
51	22.58	5.604	.1785	67.5	618.5	551.0	.1496	.1427	.2923	51
52	23.48	5.502	.1818	68.5	618.7	550.2	.1516	.1390	.2906	52
53	24.36	5.404	.1851	69.5	619.0	549.5	.1536	.1354	.2890	53
54	25.23	5.309	.1884	70.4	619.2	548.8	.1556	.1319	.2875	54
55	26.09	5.218	0.1917	71.4	619.4	548.0	0.1575	1.1284	1.2859	55
56	26.94	5.129	.1950	72.3	619.7	547.4	.1594	.1250	.2844	56
57	27.77	5.044	.1983	73.3	619.9	546.6	.1613	.1217	.2830	57
58	28.59	4.962	.2015	74.2	620.1	545.9	.1631	.1184	.2815	58
59	29.41	4.882	.2048	75.0	620.3	545.3	.1650	.1151	.2801	59
60	30.21	4.805	0.2081	75.9	620.5	544.6	0.1668	1.1119	1.2787	60
61	31.00	4.730	.2114	76.8	620.7	543.9	.1685	.1088	.2773	61
62	31.78	4.658	.2147	77.7	620.9	543.2	.1703	.1056	.2759	62
63	32.55	4.588	.2180	78.5	621.1	542.6	.1720	.1026	.2746	63
64	33.31	4.519	.2213	79.4	621.3	541.9	.1737	.0996	.2733	64
65	34.06	4.453	0.2245	80.2	621.5	541.3	0.1754	1.0966	1.2720	65
66	34.81	4.389	.2278	81.0	621.7	540.7	.1770	.0937	.2707	66
67	35.54	4.327	.2311	81.8	621.9	540.1	.1787	.0907	.2694	67
68	36.27	4.267	.2344	82.6	622.0	539.4	.1803	.0879	.2682	68
69	36.99	4.208	.2377	83.4	622.2	538.8	.1819	.0851	.2670	69
70	37.70	4.151	0.2409	84.2	622.4	538.2	0.1835	1.0823	1.2658	70
71	38.40	4.095	.2442	85.0	622.6	537.6	.1850	.0795	.2645	71
72	39.09	4.041	.2475	85.8	622.8	537.0	.1866	.0768	.2634	72
73	39.78	3.988	.2507	86.5	622.9	536.4	.1881	.0741	.2622	73
74	40.46	3.937	.2540	87.3	623.1	535.8	.1896	.0715	.2611	74
75	41.13	3.887	0.2573	88.0	623.2	535.2	0.1910	1.0689	1.2599	75
76	41.80	3.838	.2606	88.8	623.4	534.6	.1925	.0663	.2588	76
77	42.46	3.790	.2638	89.5	623.5	534.0	.1940	.0637	.2577	77
78	43.11	3.744	.2671	90.2	623.7	533.5	.1954	.0612	.2566	78
79	43.76	3.699	.2704	90.9	623.8	532.9	.1968	.0587	.2555	79
80	44.40	3.655	0.2736	91.7	624.0	532.3	0.1982	1.0563	1.2545	80

TABLE 7.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SATURATED AMMONIA: ABSOLUTE PRESSURE TABLE.—(Continued.)

Pressure (abs.), lbs./in.²	Temp. °F.	Volume vapor. ft.³/lb.	Density vapor. lbs./ft.³	Heat content. Liquid. Btu./lb.	Heat content. Vapor. Btu./lb.	Latent heat. Btu./lb.	Entropy. Liquid. Btu./lb. °F.	Entropy. Evap. Btu./lb. °F.	Entropy. Vapor. Btu./lb. °F.	Pressure (abs.), lbs./in.²
p	t	V	$1/V$	h	H	L	s	L/T	S	p
80	44.40	3.655	0.2736	91.7	624.0	532.3	0.1982	1.0563	1.2545	**80**
81	45.03	3.612	.2769	92.4	624.1	531.7	.1996	.0538	.2534	81
82	45.66	3.570	.2801	93.1	624.3	531.2	.2010	.0514	.2524	82
83	46.28	3.528	.2834	93.8	624.4	530.6	.2024	.0490	.2514	83
84	46.89	3.488	.2867	94.5	624.6	530.1	.2037	.0467	.2504	84
85	47.50	3.449	0.2899	95.1	624.7	529.6	0.2051	1.0443	1.2494	**85**
86	48.11	3.411	.2932	95.8	624.8	529.0	.2064	.0420	.2484	86
87	48.71	3.373	.2964	96.5	625.0	528.5	.2077	.0397	.2474	87
88	49.30	3.337	.2997	97.2	625.1	527.9	.2090	.0375	.2465	88
89	49.89	3.301	.3030	97.8	625.2	527.4	.2103	.0352	.2455	89
90	50.47	3.266	0.3062	98.4	625.3	526.9	0.2115	1.0330	1.2445	**90**
91	51.05	3.231	.3095	99.1	625.5	526.4	.2128	.0308	.2436	91
92	51.62	3.198	.3127	99.8	625.6	525.8	.2141	.0286	.2427	92
93	52.19	3.165	.3160	100.4	625.7	525.3	.2153	.0265	.2418	93
94	52.76	3.132	.3192	101.0	625.8	524.8	.2165	.0243	.2408	94
95	53.32	3.101	0.3225	101.6	625.9	524.3	0.2177	1.0222	1.2399	**95**
96	53.87	3.070	.3258	102.3	626.1	523.8	.2190	.0201	.2391	96
97	54.42	3.039	.3290	102.9	626.2	523.3	.2201	.0181	.2382	97
98	54.97	3.010	.3323	103.5	626.3	522.8	.2213	.0160	.2373	98
99	55.51	2.980	.3355	104.1	626.4	522.3	.2225	.0140	.2365	99
100	56.05	2.952	0.3388	104.7	626.5	521.8	0.2237	1.0119	1.2356	**100**
102	57.11	2.896	.3453	105.9	626.7	520.8	.2260	.0079	.2339	102
104	58.16	2.843	.3518	107.1	626.9	519.8	.2282	.0041	.2323	104
106	59.19	2.791	.3583	108.3	627.1	518.8	.2305	1.0002	.2307	106
108	60.21	2.741	.3648	109.4	627.3	517.9	.2327	0.9964	.2291	108
110	61.21	2.693	0.3713	110.5	627.5	517.0	0.2348	0.9927	1.2275	**110**
112	62.20	2.647	.3778	111.7	627.7	516.0	.2369	.9890	.2259	112
114	63.17	2.602	.3843	112.8	627.9	515.1	.2390	.9854	.2244	114
116	64.13	2.559	.3909	113.9	628.1	514.2	.2411	.9819	.2230	116
118	65.08	2.517	.3974	114.9	628.2	513.3	.2431	.9784	.2215	118
120	66.02	2.476	0.4039	116.0	628.4	512.4	0.2452	0.9749	1.2201	**120**
122	66.94	2.437	.4104	117.1	628.6	511.5	.2471	.9715	.2186	122
124	67.86	2.399	.4169	118.1	628.7	510.6	.2491	.9682	.2173	124
126	68.76	2.362	.4234	119.1	628.9	509.8	.2510	.9649	.2159	126
128	69.65	2.326	.4299	120.1	629.0	508.9	.2529	.9616	.2145	128
180	70.53	2.291	0.4364	121.1	629.2	508.1	0.2548	0.9584	1.2132	**180**
132	71.40	2.258	.4429	122.1	629.3	507.2	.2567	.9552	.2119	132
134	72.26	2.225	.4494	123.1	629.5	506.4	.2585	.9521	.2106	134
136	73.11	2.193	.4559	124.1	629.6	505.5	.2603	.9490	.2093	136
138	73.95	2.162	.4624	125.1	629.8	504.7	.2621	.9460	.2081	138
140	74.79	2.132	0.4690	126.0	629.9	503.9	0.2638	0.9430	1.2068	**140**
142	75.61	2.103	.4755	126.9	630.0	503.1	.2656	.9400	.2056	142
144	76.42	2.075	.4820	127.9	630.2	502.3	.2673	.9371	.2044	144
146	77.23	2.047	.4885	128.8	630.3	501.5	.2690	.9342	.2032	146
148	78.03	2.020	.4951	129.7	630.4	500.7	.2707	.9313	.2020	148
150	78.81	1.994	0.5016	130.6	630.5	499.9	0.2724	0.9285	1.2009	**150**

TABLE 7.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SATURATED AMMONIA: ABSOLUTE PRESSURE TABLE.—(Concluded.)

Pressure (abs.). lbs./in.²	Temp. °F.	Volume vapor. ft.³/lb.	Density vapor. lbs./ft.³	Heat content. Liquid. Btu./lb.	Heat content. Vapor. Btu./lb.	Latent heat. Btu./lb.	Entropy. Liquid. Btu./lb. °F.	Entropy. Evap. Btu./lb. °F.	Entropy. Vapor. Btu./lb. °F.	Pressure (abs.). lbs./in.²
p	t	V	$1/V$	h	H	L	s	L/T	S	p
150	78.81	1.994	0.5016	130.6	630.5	499.9	0.2724	0.9285	1.2009	**150**
152	79.60	1.968	.5081	131.5	630.6	499.1	.2740	.9257	.1997	152
154	80.37	1.943	.5147	132.4	630.7	498.3	.2756	.9229	.1985	154
156	81.13	1.919	.5212	133.3	630.9	497.6	.2772	.9202	.1974	156
158	81.89	1.895	.5277	134.2	631.0	496.8	.2788	.9175	.1963	158
160	82.04	1.872	0.5343	135.0	631.1	496.1	0.2804	0.9148	1.1952	**160**
162	83.39	1.849	.5408	135.9	631.2	495.3	.2820	.9122	.1942	162
164	84.12	1.827	.5473	136.8	631.3	494.5	.2835	.9096	.1931	164
166	84.85	1.805	.5539	137.6	631.4	493.8	.2850	.9070	.1920	166
168	85.57	1.784	.5604	138.4	631.5	493.1	.2866	.9044	.1910	168
170	86.29	1.764	0.5670	139.3	631.6	492.3	0.2881	0.9019	1.1900	**170**
172	87.00	1.744	.5735	140.1	631.7	491.6	.2895	.8994	.1889	172
174	87.71	1.724	.5801	140.9	631.7	490.8	.2910	.8969	.1879	174
176	88.40	1.705	.5866	141.7	631.8	490.1	.2925	.8944	.1869	176
178	89.10	1.686	.5932	142.5	631.9	489.4	.2939	.8920	.1859	178
180	89.78	1.667	0.5998	143.3	632.0	488.7	0.2954	0.8896	1.1850	**180**
182	90.46	1.649	.6063	144.1	632.1	488.0	.2968	.8872	.1840	182
184	91.14	1.632	.6129	144.8	632.1	487.3	.2982	.8848	.1830	184
186	91.80	1.614	.6195	145.6	632.2	486.6	.2996	.8825	.1821	186
188	92.47	1.597	.6261	146.4	632.3	485.9	.3010	.8801	.1811	188
190	93.13	1.581	0.6326	147.2	632.4	485.2	0.3024	0.8778	1.1802	**190**
192	93.78	1.564	.6392	147.9	632.4	484.5	.3037	.8755	.1792	192
194	94.43	1.548	.6458	148.7	632.5	483.8	.3050	.8733	.1783	194
196	95.07	1.533	.6524	149.5	632.6	483.1	.3064	.8710	.1774	196
198	95.71	1.517	.6590	150.2	632.6	482.4	.3077	.8688	.1765	198
200	96.34	1.502	0.6656	150.9	632.7	481.8	0.3090	0.8666	1.1756	**200**
205	97.90	1.466	.6821	152.7	632.8	480.1	.3122	.8612	.1734	205
210	99.43	1.431	.6986	154.6	633.0	478.4	.3154	.8559	.1713	210
215	100.94	1.398	.7152	156.3	633.1	476.8	.3185	.8507	.1692	215
220	102.42	1.367	.7318	158.0	633.2	475.2	.3216	.8455	.1671	220
225	103.87	1.336	0.7484	159.7	633.3	473.6	0.3246	0.8405	1.1651	**225**
230	105.30	1.307	.7650	161.4	633.4	472.0	.3275	.8356	.1631	230
235	106.71	1.279	.7817	163.1	633.5	470.4	.3304	.8307	.1611	235
240	108.09	1.253	.7984	164.7	633.6	468.9	.3332	.8260	.1592	240
245	109.46	1.227	.8151	166.4	633.7	467.3	.3360	.8213	.1573	245
250	110.80	1.202	0.8319	168.0	633.8	465.8	0.3388	0.8167	1.1555	**250**
255	112.12	1.178	.8487	169.5	633.8	464.3	.3415	.8121	.1536	255
260	113.42	1.155	.8655	171.1	633.9	462.8	.3441	.8077	.1518	260
265	114.71	1.133	.8824	172.6	633.9	461.3	.3468	.8033	.1501	265
270	115.97	1.112	.8993	174.1	633.9	459.8	.3494	.7989	.1483	270
275	117.22	1.091	0.9162	175.6	634.0	458.4	0.3519	0.7947	1.1466	**275**
280	118.45	1.072	.9332	177.1	634.0	456.9	.3545	.7904	.1449	280
285	119.66	1.052	.9502	178.6	634.0	455.4	.3569	.7863	.1432	285
290	120.86	1.034	.9672	180.0	634.0	454.0	.3594	.7821	.1415	290
295	122.05	1.016	.9843	181.5	634.0	452.5	.3618	.7781	.1399	295
300	123.21	0.999	1.0015	182.9	634.0	451.1	0.3642	0.7741	1.1383	**300**

TABLE 8.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SATURATED AMMONIA: GAUGE PRESSURE TABLE.

Pressure. (gage). lbs./in.²	Temp. °F.	Volume vapor. ft.³/lb.	Density vapor. lbs./ft.³	Heat content. Liquid. Btu./lb.	Heat content. Vapor. Btu./lb.	Latent heat. Btu./lb.	Entropy. Liquid. Btu./lb.°F.	Entropy. Evap. Btu./lb.°F.	Entropy. Vapor. Btu./lb.°F.	Pressure (gage). lbs./in.²
g. p.	*t*	*V*	*1/V*	*h*	*H*	*L*	*s*	*L/T*	*S*	*g. p.*
20*	−63.9	50.5	0.0198	−25.3	588.0	613.3	−0.062	1.550	1.488	20*
19*	−61.0	46.2	.0217	−22.3	589.2	611.5	− .055	.535	.480	19*
18*	−58.4	42.6	.0235	−19.5	590.3	609.8	− .048	.521	.473	18*
17*	−55.9	39.5	.0253	−16.9	591.3	608.2	− .041	.507	.466	17*
16*	−53.6	36.8	.0272	−14.5	592.2	606.7	− .035	.495	.460	16*
15*	−51.4	34.5	0.0290	−12.2	593.1	605.3	−0.029	1.483	1.454	15*
14*	−49.4	32.5	.0308	−10.0	593.9	603.9	− .023	.472	.449	14*
13*	−47.4	30.7	.0326	− 7.9	594.7	602.6	− .019	.462	.443	13*
12*	−45.6	29.1	.0344	− 5.9	595.4	601.3	− .014	.452	.438	12*
11*	−43.8	27.6	.0362	− 4.0	596.1	600.1	− .010	.443	.433	11*
10*	−42.1	26.3	0.0380	− 2.2	596.8	599.0	−0.005	1.434	1.429	10*
9*	−40.4	25.2	.0397	− 0.5	597.4	597.9	− .001	.426	.425	9*
8*	−38.9	24.1	.0415	+ 1.2	598.0	596.8	+ .003	.418	.421	8*
7*	−37.3	23.1	.0433	2.8	598.6	595.8	.007	.411	.418	7*
6*	−35.9	22.2	.0450	4.4	599.1	594.7	.010	.405	.415	6*
5*	−34.5	21.4	0.0468	5.9	599.6	593.7	0.014	1.397	1.411	5*
4*	−33.1	20.6	.0485	7.4	600.2	592.8	.017	.390	.407	4*
3*	−31.8	19.9	.0503	8.8	600.7	591.9	.020	.384	.404	3*
2*	−30.5	19.2	.0520	10.2	601.2	591.0	.024	.377	.401	2*
1*	−29.2	18.6	.0538	11.5	601.6	590.1	.027	.371	.398	1*
0	−28.0	18.0	0.0555	12.8	602.1	589.3	0.030	1.366	1.396	0
1	−25.6	16.9	.0590	15.4	603.0	587.6	.036	.354	.390	1
2	−23.4	16.0	.0626	17.8	603.8	586.0	.041	.344	.385	2
3	−21.2	15.1	.0661	20.1	604.6	584.5	.047	.333	.380	3
4	−19.2	14.4	.0695	22.3	605.3	583.0	.052	.324	.376	4
5	−17.2	13.7	0.0730	24.4	606.0	581.6	0.056	1.315	1.371	5
6	−15.3	13.1	.0765	26.4	606.6	580.2	.061	.306	.367	6
7	−13.5	12.5	.0799	28.4	607.3	578.9	.065	.298	.363	7
8	−11.8	12.0	.0834	30.3	607.9	577.6	.070	.290	.360	8
9	−10.1	11.5	.0868	32.1	608.4	576.3	.074	.282	.356	9
10	− 8.4	11.1	0.0902	33.8	609.0	575.2	0.078	1.275	1.353	10
11	− 6.9	10.7	.0937	35.5	609.5	574.0	.081	.268	.349	11
12	− 5.3	10.3	.0971	37.1	610.0	572.9	.085	.261	.346	12
13	− 3.8	9.96	.100	38.8	610.5	571.7	.088	.255	.343	13
14	− 2.4	9.63	.104	40.4	611.0	570.6	.092	.248	.340	14
15	− 1.0	9.32	0.107	41.9	611.4	569.5	0.095	1.242	1.337	15
16	+ 0.4	9.04	.111	43.4	611.9	568.5	.098	236	.334	16
17	1.7	8.78	.114	44.8	612.3	567.5	.101	.230	.331	17
18	3.0	8.53	.117	46.2	612.7	566.5	.104	.225	.329	18
19	4.3	8.28	.121	47.6	613.1	565.5	.107	.219	.326	19
20	5.5	8.06	0.124	48.9	613.5	564.6	0.110	1.214	1.324	20
21	6.7	7.85	.127	50.2	613.9	563.7	.113	209	.322	21
22	7.9	7.65	.131	51.5	614.2	562.7	.116	.204	.320	22
23	9.1	7.46	.134	52.8	614.6	561.8	.119	199	.318	23
24	10.2	7.28	.138	54.0	614.9	560.9	121	.194	315	24
25	11.3	7.11	0.141	55.3	615.3	560.0	0.124	1.189	1.313	25
26	12.4	6.94	.144	56.5	615.6	559.1	.126	.185	.311	26
27	13.5	6.78	.148	57.6	615.9	558.3	.129	.180	.309	27
28	14.5	6.63	.151	58.8	616.2	557.4	131	.176	307	28
29	15.6	6.49	.154	59.9	616.5	556.6	.134	.171	305	29

* Inches of mercury below one standard atmosphere (29.92 in.).

TABLE 8.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SATURATED AMMONIA: GAUGE PRESSURE TABLE.—(Continued)

Pressure (gage), lbs./in.²	Temp. °F.	Volume vapor, ft.³/lb.	Density vapor, lbs./ft.³	Heat content. Liquid, Btu./lb.	Heat content. Vapor, Btu./lb.	Latent heat, Btu./lb.	Entropy. Liquid, Btu./lb. °F.	Entropy. Evap., Btu./lb. °F.	Entropy. Vapor, Btu./lb. °F.	Pressure (gage), lbs./in.²
g. p.	t	V	1/V	h	H	L	s	L/T	S	g. p.
30	16.6	6.35	0.158	61.0	616.8	555.8	0.136	1.167	1.303	30
31	17.6	6.22	.161	62.1	617.1	555.0	.138	.163	.301	31
32	18.6	6.09	.164	63.2	617.4	554.2	.140	.159	.299	32
32	19.5	5.97	.168	64.2	617.6	553.4	.143	.155	.298	33
34	20.5	5.85	.171	65.3	617.9	552.6	.145	.151	.296	34
35	21.4	5.74	0.174	66.3	618.2	551.9	0.147	1.148	1.295	35
36	22.3	5.64	.177	67.3	618.4	551.1	.149	.144	.293	36
37	23.2	5.54	.181	68.3	618.7	550.4	.151	.140	.291	37
38	24.1	5.44	.184	69.2	618.9	549.7	.153	.137	.290	38
39	25.0	5.34	.187	70.2	619.1	548.9	.155	.133	.288	39
40	25.8	5.25	0.191	71.2	619.4	548.2	0.157	1.130	1.287	40
41	26.7	5.16	.194	72.1	619.6	547.5	.159	.126	.285	41
42	27.5	5.07	.197	73.0	619.8	546.8	.161	.123	.284	24
43	28.3	4.99	.201	73.9	620.0	546.1	.163	.119	.282	43
44	29.2	4.91	.204	74.8	620.3	545.5	.164	.116	.280	44
45	30.0	4.83	0.207	75.7	620.5	544.8	0.166	1.113	1.279	45
46	30.8	4.76	.210	76.6	620.7	544.1	.168	.110	.278	46
47	31.5	4.68	.214	77.4	62).9	543.5	.170	.107	.277	47
48	32.3	4.61	.217	78.3	621.1	542.8	.171	.104	.275	48
49	33.1	4.54	.220	79.1	621.3	542.2	.173	.101	.274	49
50	33.8	4.48	0.224	80.0	621.5	541.5	0.175	1.098	1.273	50
51	34.6	4.41	.227	80.8	621.7	540.9	.177	.095	.272	51
52	35.3	4.35	.230	81.6	621.8	540.2	.178	.092	.270	52
53	36.1	4.29	.233	82.4	622.0	539.6	.180	.089	.269	53
54	36.8	4.23	.237	83.2	622.2	539.0	.181	.086	.267	54
55	37.5	4.17	0.240	84.0	622.4	538.4	0.183	1.083	1.266	55
56	38.2	4.12	.243	84.8	622.5	537.7	.185	.080	.265	56
57	38.9	4.06	.246	85.6	622.7	537.1	.186	.078	.264	57
58	39.6	4.01	.250	86.3	622.9	536.6	.188	.075	.263	58
59	40.3	3.96	.253	87.0	623.0	536.0	.189	.072	.261	59
60	40.9	3.91	0.256	87.8	623.2	535.4	0.191	1.069	1.260	60
61	41.6	3.86	.260	88.6	623.4	534.8	.192	.067	.259	61
62	42.3	3.81	.263	89.3	623.5	534.2	.194	.064	.258	62
63	42.9	3.77	.266	90.0	623.7	533.7	.195	.062	.257	63
64	43.6	3.72	.269	90.7	623.8	533.1	.196	.060	.256	64
65	44.2	3.67	0.273	91.5	624.0	532.5	0.198	1.057	1.255	65
66	44.8	3.63	.276	92.2	624.1	531.9	.199	.055	.254	66
67	45.5	3.59	.279	92.9	624.2	531.3	.201	.052	.253	67
68	46.1	3.55	.282	93.6	624.4	530.8	.202	.050	.252	68
69	46.7	3.51	.286	94.3	624.5	530.2	.203	.048	.251	69
70	47.3	3.47	0.289	94.9	624.6	529.7	0.205	1.045	1.250	70
71	47.9	3.43	.292	95.6	624.8	529.2	.206	.043	.249	71
72	48.5	3.39	.295	96.3	624.9	528.6	.207	.041	.248	72
73	49.1	3.35	.299	97.0	625.1	528.1	.209	.038	.247	73
74	49.7	3.32	.302	97.6	625.2	527.6	.210	.036	.246	74
75	50.3	3.28	0.305	98.3	625.3	527.0	0.211	1.034	1.245	75
76	50.9	3.24	.308	98.9	625.4	526.5	.212	.032	.244	76
77	51.5	3.21	.312	99.5	625.5	526.0	.214	.029	.243	77
78	52.0	3.17	.315	100.2	625.7	525.5	.215	.027	.242	78
79	52.6	3.14	.318	100.8	625.8	525.0	.216	.025	.241	79

TABLE 8.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SATURATED AMMONIA: GAUGE PRESSURE TABLE.—(Continued)

Pressure (gage). lbs./in.	Temp. °F.	Volume vapor. ft.³/lb.	Density vapor. lbs./ft.³	Heat content. Liquid. Btu./lb.	Heat content. Vapor. Btu./lb.	Latent heat. Btu./lb.	Entropy. Liquid Btu./lb. °F.	Entropy. Evap. Btu./lb. °F.	Entropy. Vapor. Btu./lb. °F.	Pressure (gage). lbs./in.
g. p.	t	V	1/V	h	H	L	s	L/T	S	g. p.
80	53.1	3.11	0.322	101.5	625.9	524.4	0.217	1.023	1.240	**80**
81	53.7	3.08	.325	102.1	626.0	523.9	.219	020	239	81
82	54.3	3.05	328	102.7	626.1	523.4	220	018	238	82
83	54.8	3.02	331	103.3	626.3	523.0	.221	016	.237	83
84	55.3	2.99	335	103.9	626.4	522.5	222	015	.237	84
85	55.9	2.96	0.338	104.5	626.5	522.0	0.223	1.013	1.236	**85**
86	56.4	2.94	.341	105.1	626.6	521.5	.224	011	.235	86
87	57.0	2.91	344	105.7	626.7	521.0	226	008	.234	87
88	57.5	2.88	348	106.3	626.8	520.5	227	006	.233	88
89	58.0	2.85	351	106.9	626.9	520.0	.228	005	.233	89
90	58.5	2.82	0.354	107.5	627.0	519.5	0.229	1.003	1.232	**90**
91	59.0	2.80	357	108.1	627.1	519.0	.230	1.001	.231	91
92	59.6	2.77	361	108.7	627.2	518.5	.231	0.999	.230	92
93	60.1	2.75	364	109.3	627.3	518.0	.232	997	.229	93
94	60.6	2.72	367	109.8	627.4	517.6	233	995	.228	94
95	61.1	2.70	0.370	110.4	627.5	517.1	0.235	0.993	1.228	**95**
96	61.6	2.68	374	111.0	627.6	516.6	.236	991	.227	96
97	62.0	2.65	377	111.6	627.7	516.1	237	.989	.226	97
98	62.5	2.63	380	112.1	627.8	515.7	238	.988	.226	98
99	63.0	2.61	383	112.6	627.9	515.3	.239	.986	.225	99
100	63.5	2.59	0.287	113.2	628.0	514.8	0.240	0.984	1.224	**100**
102	64.5	2.54	393	114.2	628.1	513.9	242	981	.223	102
104	65.4	2.50	400	115.3	628.3	513.0	244	977	.221	104
106	66.4	2.46	406	116.4	628.5	512.1	246	974	220	106
108	67.3	2.42	.413	117.4	628.6	511.2	248	.970	218	108
110	68.2	2.39	0.419	118.5	628.8	510.3	0.250	0.967	1.217	**110**
112	69.1	2.35	426	119.5	628.9	509.4	.252	.964	.216	112
114	70.0	2.31	432	120.5	629.1	508.6	.254	.960	.214	114
116	70.8	2.28	439	121.5	629.3	507.8	.256	.957	.213	116
118	71.7	2.25	445	122.5	629.4	506.9	.257	.954	211	118
120	72.6	2.21	0.452	123.5	629.5	506.0	0.259	0.951	1.210	**120**
122	73.4	2.18	458	124.5	629.7	505.2	.261	.948	.209	122
124	74.2	2.15	465	125.4	629.8	504.4	263	.945	208	124
126	75.1	2.12	471	126.3	629.9	503.6	.364	.942	.206	126
128	75.9	2.09	.478	127.3	630.1	502.8	.266	.939	205	128
130	76.7	2.06	0.484	128.2	630.2	502.0	0.268	0.936	1.204	**130**
132	77.5	2.04	.491	129.1	630.3	501.2	270	.933	.203	132
134	78.3	2.01	.497	130.0	630.4	500.4	271	930	201	134
136	79.1	1.98	.504	130.9	630.5	499.6	.273	.927	200	136
138	79.9	1.96	.510	131.8	630.7	498.9	.274	925	.199	138
140	80.6	1.93	0.517	132.7	630.8	498.1	0.276	0.922	1.198	**140**
142	81.4	1.91	.523	133.6	630.9	497.3	.278	919	.197	142
144	82.2	1.89	530	134.5	631.0	496.5	.279	.917	.196	144
146	82.9	1.86	536	135.3	631.1	495.8	.281	.914	195	146
148	83.6	1.84	.543	136.2	631.2	495.0	283	.911	194	148

TABLE 8.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SATURATED AMMONIA: GAUGE PRESSURE TABLE.—(Concluded.)

Pressure (gage). lbs./in.	Temp. °F.	Volume vapor. ft.³/lb.	Density vapor. lbs./ft.³	Heat content. Liquid. Btu./lb.	Heat content. Vapor. Btu./lb.	Latent heat. Btu./lb.	Entropy. Liquid Btu./lb. °F.	Entropy. Evap. Btu./lb. °F.	Entropy. Vapor. Btu./lb. °F.	Pressure (gage). lbs./in.
$g.p.$	t	V	$1/V$	h	H	L	s	L/T	S	$g.p.$
150	84.4	1.82	0.550	137.0	631.3	494.3	0.284	0.909	1.193	**150**
152	85.1	1.80	.556	137.9	631.4	493.5	.286	.906	.192	152
154	85.8	1.78	.563	138.7	631.5	492.8	.287	.904	.191	154
156	86.5	1.76	.569	139.5	631.6	492.1	.289	.901	.190	156
158	87.2	1.74	.576	140.3	631.7	491.4	.290	.899	.189	158
160	88.0	1.72	0.582	141.1	631.8	490.7	0.292	0.896	1.188	**160**
162	88.6	1.70	.589	141.9	631.9	490.0	.293	.894	.187	162
164	89.3	1.68	.595	142.7	631.9	489.2	.294	.891	185	164
166	90.0	1.66	.602	143.5	632.0	488.5	.296	889	.185	166
168	90.7	1.64	.609	144.3	632.1	487.8	.297	886	.183	168
170	91.4	1.62	0.615	145.1	632.1	487.0	0.299	0.884	1.183	**170**
172	92.0	1.61	.622	145.8	632.2	486.4	.300	882	182	172
174	92.7	1.59	.628	146.6	632.3	485.7	.302	879	181	174
176	93.4	1.57	.635	147.4	632.4	485.0	.303	.877	.180	176
178	94.0	1.56	.641	148.2	632.5	484.3	.304	.875	.179	178
180	94.7	1.54	0.648	148.9	632.5	483.6	0.305	0.873	1.178	**180**
182	95.3	1.53	.655	149.7	632.6	482.9	.307	.870	.177	182
184	95.9	1.51	.661	150.5	632.7	482.2	.308	.868	.176	184
186	96.6	1.50	.668	151.2	632.7	481.5	.309	.866	.175	186
188	97.2	1.48	.674	151.9	632.8	480.9	.311	.863	.174	188
190	97.8	1.47	0.681	152.6	632.8	480.2	0.312	0.861	1.173	**190**
192	98.4	1.45	.688	153.4	632.9	479.5	.314	859	.173	192
194	99.0	1.44	.694	154.0	632.9	478.9	.315	.857	.172	194
196	99.7	1.43	701	154.8	633.0	478.2	.316	.855	171	196
198	100.3	1.41	708	155.5	633.0	477.5	.317	.853	.170	198
200	100.9	1.40	0.714	156.2	633.1	476.9	0.318	0.851	1.169	**200**
205	102.3	1.37	.731	158.0	633.2	475.2	.321	.846	167	205
210	103.8	1.34	.747	159.6	633.3	473.7	.324	.841	.165	210
215	105.2	1.31	.764	161.3	633.4	472.1	.327	.836	.163	215
220	106.6	1.28	.781	163.0	633.5	470.5	.330	.831	.161	220
225	108.0	1.25	0.797	164.6	633.6	469.0	0.333	0.826	1.159	**225**
230	109.4	1.23	.814	166.3	633.7	467.4	.336	822	.158	230
235	110.7	1.20	.831	167.9	633.8	465.9	.339	.817	156	235
240	112.0	1.18	.848	169.4	633.8	464.4	341	.813	154	240
245	113.3	1.16	.864	171.0	633.9	462.9	.344	.808	152	245
250	114.6	1.13	0.881	172.6	633.9	461.3	0.346	0.804	1.150	**250**
255	115.9	1.11	.898	174.1	634.0	459.9	349	.799	148	255
260	117.1	1.09	.915	175.6	634.0	458.4	.352	795	147	260
265	118.4	1.07	.932	177.0	634.0	457.0	.354	791	.145	265
270	119.6	1.05	.949	178.5	634.0	455.5	.357	.788	.143	270
275	120.8	1.03	0.966	179.9	634.0	454.1	0.359	0.783	1.142	**275**
280	122.0	1.02	.983	181.4	634.0	452.6	.362	.778	140	280
285	123.1	1.00	1.000	182.8	634.0	451.2	.364	.774	.138	285
290	124.3	0.98	1.018	184.2	634.0	449.8	367	.770	137	290
295	125.4	0.97	1.035	185.6	634.0	448.4	.369	766	135	295
300	126.5	0.95	1.052	187.0	633.9	446.9	0.371	0.762	1.133	**300**

TABLE 9.—PROPERTIES OF LIQUID AMMONIA.

Temp. °F. t	Saturation. Pressure (abs.), lbs./in.² p	Volume. ft.³/lb. v	Density. lbs./ft.³ $\frac{1}{v}$	Specific heat. Btu./lb. °F. c	Heat content. Btu./lb. h	Latent heat. Btu./lb. L	Latent heat of pressure variation. $\frac{\text{Btu./lb.}}{\text{lb./in.}^2}$ l	Variation of h with p (t constant). $\frac{\text{Btu./lb.}}{\text{lb./in.}^2}$ $\left(\frac{\partial h}{\partial p}\right)_t$	Compressibility. per lb./in.²×10⁶ $-\frac{1}{v}\left(\frac{\partial v}{\partial p}\right)_t$	Temp. °F. t
Triple point.	......	0.01961*	51.00*	......	......	......	......	......	......	......
	0.88	.02182	45.83	......	......	......	......	......	......	−107.86
−100	1.24	0.02197	45.52	(1.040)	(−63.0)	(633)	......	......	......	−100
−95	1.52	.02207	45.32	(1.042)	(−57.8)	(631)	......	......	......	−95
−90	1.86	.02216	45.12	(1.043)	(−52.6)	(628)	......	......	......	−90
−85	2.27	.02226	44.92	(1.045)	(−47.4)	(625)	......	......	......	−85
−80	2.74	.02236	44.72	(1.046)	(−42.2)	(622)	......	......	......	−80
−75	3.29	0.02246	44.52	(1.048)	(−36.9)	(619)	......	......	......	−75
−70	3.94	.02256	44.32	(1.050)	(−31.7)	(616)	......	......	......	−70
−65	4.69	.02267	44.11	(1.052)	(−26.4)	(613)	......	......	......	−65
−60	5.55	.02278	43.91	1.054	−21.18	610.8	−0.0016	0.0026	4.4	−60
−55	6.54	.02288	43.70	1.056	−15.90	607.5	−.0016	.0026	4.5	−55
−50	7.67	0.02299	43.49	1.058	−10.61	604.3	−0.0017	0.0026	4.6	−50
−45	8.95	.02310	43.28	1.060	−5.31	600.9	−.0017	.0026	4.7	−45
−40	10.41	.02322	43.08	1.062	0.00	597.6	−.0018	.0025	4.8	−40
−35	12.05	.02333	42.86	1.064	+5.32	594.2	−.0018	.0025	5.0	−35
−30	13.90	.02345	42.65	1.066	10.66	590.7	−.0019	.0025	5.1	−30
−25	15.98	0.02357	42.44	1.068	16.00	587.2	−0.0019	0.0024	5.2	−25
−20	18.30	.02369	42.22	1.070	21.36	583.6	−.0020	.0024	5.4	−20
−15	20.88	.02381	42.00	1.073	26.73	580.0	−.0020	.0024	5.5	−15
−10	23.74	.02393	41.78	1.075	32.11	576.4	−.0021	.0023	5.7	−10
−5	26.92	.02406	41.56	1.078	37.51	572.6	−.0022	.0023	5.8	−5
0	30.42	0.02419	41.34	1.080	42.92	568.9	−0.0022	0.0022	6.0	0
5	34.27	.02432	41.11	1.083	48.35	565.0	−.0023	.0022	6.2	5
10	38.51	.02446	40.89	1.085	53.79	561.1	−.0024	.0021	6.4	10
15	43.14	.02460	40.66	1.088	59.24	557.1	−.0025	.0021	6.6	15
20	48.21	.02474	40.43	1.091	64.71	553.1	−.0025	.0020	6.8	20
25	53.73	0.02488	40.20	1.094	70.20	548.9	−0.0026	0.0020	7.0	25
30	59.74	.02503	39.96	1.097	75.71	544.8	−.0027	.0019	7.3	30
35	66.26	.02518	39.72	1.100	81.23	540.5	−.0028	.0019	7.5	35
40	73.32	.02533	39.49	1.104	86.77	536.2	−.0029	.0018	7.8	40
45	80.96	.02548	39.24	1.108	92.34	531.8	−.0030	.0017	8.1	45
50	89.19	0.02564	39.00	1.112	97.93	527.3	−0.0031	0.0017	8.4	50
55	98.06	.02581	38.75	1.116	103.54	522.8	−.0032	.0016	8.8	55
60	107.6	.02597	38.50	1.120	109.18	518.1	−.0033	.0015	9.1	60
65	117.8	.02614	38.25	1.125	114.85	513.4	−.0034	.0014	9.5	65
70	128.8	.02632	38.00	1.129	120.54	508.6	−.0035	.0013	10.0	70
75	140.5	0.02650	37.74	1.133	126.25	503.7	−0.0037	0.0012	10.4	75

*Properties of solid ammonia at the triple point (−107.86 °F.)

Temp. °F. t	Saturation. Pressure (abs.), lbs./in.² p	Volume. ft.³/lb. v	Density. lbs./ft.³ $\frac{1}{v}$	Specific heat. Btu./lb. °F. c	Heat content. Btu./lb. h	Latent heat. Btu./lb. L	Latent heat of pressure variation. $\frac{\text{Btu./lb.}}{\text{lb./in.}^2}$ l	Variation of h with p (t constant). $\frac{\text{Btu./lb.}}{\text{lb./in.}^2}$ $\left(\frac{\partial h}{\partial p}\right)_t$	Compressibility. per lb./in.²×10⁶ $-\frac{1}{v}\left(\frac{\partial v}{\partial p}\right)_t$	Temp. °F. t
75	140.5	0.02650	37.74	1.133	126.25	503.7	−0.0037	0.0012	10.4	75
80	153.0	.02668	37.48	1.138	131.99	498.7	−.0038	0011	10.9	80
85	166.4	02687	37.21	1.142	137.75	493.6	−.0040	.0010	11.4	85
90	180.6	02707	36.95	1.147	143.54	488.5	−.0041	.0009	12.0	90
95	195.8	02727	36.67	1.151	149.36	483.2	− 0043	.0008	12.6	95
100	211.9	0.02747	36.40	1.156	155.21	477.8	−0.0045	0.0006	13.3	100
105	228.9	.02769	36.12	1.162	161.09	472.3	−.0047	.0005	14.1	105
110	247.0	.02790	35.84	1.168	167.01	466.7	− 0049	0003	14.9	110
115	266.2	.02813	35.55	1.176	172.97	460.9	−.0051	.0001	15.8	115
120	286.4	.02836	35.26	1.183	178.98	455.0	−.0053	.0000	16.7	120
125	307.8	0.02860	34.96	(1.189)	(185)	(449)	......	......	......	125
130	330.3	.02885	34.66	(1.197)	(191)	(443)	......	......	......	130
135	354.1	.02911	34.35	(1.205)	(197)	(436)	......	......	......	135
140	379.1	.02938	34.04	(1.213)	(203)	(430)	......	......	......	140
145	405.5	.02966	33.72	(1.222)	(210)	(423)	......	......	......	145
150	433.2	0.02995	33.39	(1.23)	(216)	(416)	......	......	......	150
155	462.3	.03025	33.06	(1.24)	(222)	(409)	......	......	......	155
160	492.8	.03056	32.72	(1.25)	(229)	(401)	......	......	......	160
165	524.8	.03089	32.37	(1.26)	(235)	(394)	......	......	......	165
170	558.4	03124	32.01	(1.27)	(241)	(386)	......	......	......	170
175	593.5	0.03160	31.65	(1.29)	(248)	(377)	......	......	......	175
180	630.3	03198	31.27	(1.30)	(255)	(369)	......	......	......	180
185	668.7	.03238	30.88	(1.32)	(262)	(360)	......	......	......	185
190	708.9	.03281	30.48	(1.34)	(269)	(351)	......	......	......	190
195	750.9	03326	30.06	(1.36)	(276)	(342)	......	......	......	195
200	794.7	0.03375	29.63	(1.38)	(283)	(332)	......	......	......	200
210	888.1	.03482	28.72	(1.43)	(297)	(310)	......	......	......	210
220	989.5	.0361	27.7	(1.49)	(313)	(287)	......	......	......	220
230	1099.5	.0376	26.6	(1.57)	(329)	(260)	......	......	......	230
240	1218.5	.0395	25.3	(1.70)	(346)	(229)	......	......	......	240
250	1347	.0422	23.7	(1.90)	(365)	(192)	......	......	......	250
260	1486	.0463	21.6	(2.33)	(387)	(142)	......	......	......	260
270	1635	0577	17.3	(5.30)	(419)	(52)	......	......	......	270
Critical	1657	.0686	14.6	∞	(433)	0	−∞	−∞	∞	271.4

NOTE.—The figures in parentheses were calculated from empirical equations given in Bureau of Standards Scientific Papers Nos. 313 and 315 and represent values obtained by extrapolation beyond the range covered in the experimental work.

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.

(V = volume in ft.^{3}/lb; H = heat content in Btu./lb;
S = entropy in Btu./lb °F.)

Temp. °F.	Absolute pressure in lbs./in.^{2} (Saturation temperature in italics.)									Temp. °F.
	5 *−63.11°*			6 *−57.64°*			7 *−52.88°*			
	V	H	S	V	H	S	V	H	S	
Sat.	*49.31*	*588.3*	*1.4857*	*41.59*	*590.6*	*1.4708*	*36.01*	*592.6*	*1.4574*	Sat.
−50	51.05	595.2	1.5025	42.44	594.6	1.4803	36.29	594.0	1.4611	−50
−40	52.36	600.3	.5149	43.55	599.8	.4928	37.25	599.3	.4739	−40
−30	53.67	605.4	.5269	44.64	604.9	.5049	38.19	604.5	.4861	−30
−20	54.97	610.4	.5386	45.73	610.0	.5166	39.13	609.6	.4979	−20
−10	56.26	615.4	.5498	46.82	615.1	.5280	40.07	614.7	.5094	−10
0	57.55	620.4	1.5608	47.90	620.1	1.5391	41.00	619.8	1.5206	0
10	58.84	625.4	.5716	48.98	625.2	.5499	41.93	624.9	.5314	10
20	60.12	630.4	.5821	50.05	630.2	.5605	42.85	629.9	.5421	20
30	61.41	635.4	.5925	51.12	635.2	.5708	43.77	635.0	.5525	30
40	62.69	640.4	.6026	52.19	640.2	.5810	44.69	640.0	.5627	40
50	63.96	645.5	1.6125	53.26	645.2	1.5910	45.61	645.0	1.5727	50
60	65.24	650.5	.6223	54.32	650.3	.6008	46.53	650.1	.5825	60
70	66.51	655.5	.6319	55.39	655.3	.6104	47.44	655.2	.5921	70
80	67.79	660.6	.6413	56.45	660.4	.6199	48.36	660.2	.6016	80
90	69.06	665.6	.6506	57.51	665.5	.6292	49.27	665.3	.6110	90
100	70.33	670.7	1.6598	58.58	670.6	1.6384	50.18	670.4	1.6202	100
110	71.60	675.8	.6689	59.64	675.7	.6474	51.09	675.5	.6292	110
120	72.87	680.9	.6778	60.70	680.8	.6563	52.00	680.7	.6382	120
130	74.14	686.1	.6865	61.76	685.9	.6651	52.91	685.8	.6470	130
140	75.41	691.2	.6952	62.82	691.1	.6738	53.82	691.0	.6557	140
150	76.68	696.4	.7038	63.87	696.3	1.6824	54.73	696.2	1.6643	150
160	77.95	701.6	.7122	64.93	701.5	.6909	55.63	701.4	.6727	160
170	79.21	706.8	.7206	65.99	706.7	.6992	56.54	706.6	.6811	170
180	80.48	712.1	.7289	67.05	712.0	.7075	57.45	711.9	.6894	180
	10 *−41.34°*			11 *−38.14°*			12 *−35.16°*			
Sat.	*25.81*	*597.1*	*1.4276*	*23.61*	*598.3*	*1.4196*	*21.77*	*599.4*	*1.4124*	Sat.
−30	26.58	603.2	1.4420	24.12	602.7	1.4300	22.07	602.3	1.4190	−30
−20	27.26	608.5	.4542	24.74	608.1	.4423	22.64	607.7	.4314	−20
−10	27.92	613.7	.4659	25.35	613.3	.4542	23.20	613.0	.4434	−10
0	28.58	618.9	1.4773	25.95	618.5	1.4656	23.75	618.2	1.4549	0
10	29.24	624.0	.4884	26.55	623.7	.4768	24.31	623.4	.4661	10
20	29.90	629.1	.4992	27.15	628.9	.4876	24.86	628.6	4770	20
30	30.55	634.2	.5097	27.74	634.0	.4982	25.41	633.7	.4877	30
40	31.20	639.3	.5200	28.34	639.1	.5085	25.95	638.9	.4980	40
50	31.85	644.4	1.5301	28.93	644.2	1.5187	26.49	644.0	1.5082	50
60	32.49	649.5	.5400	29.52	649.3	.5286	27.03	649.1	.5182	60
70	33.14	654.6	.5497	30.10	654.4	.5383	27.57	654.3	.5279	70
80	33.78	659.7	.5593	30.69	659.6	.5479	28.11	659.4	.5375	80
90	34.42	664.8	.5687	31.28	664.7	.5573	28.65	664.5	.5470	90
100	35.07	670.0	1.5779	31.86	669.8	1.5666	29.19	669.7	1.5562	100
110	35.71	675.1	.5870	32.44	675.0	.5757	29.72	674.8	.5654	110
120	36.35	680.3	.5960	33.03	680.1	.5847	30.26	680.0	.5744	120
130	36.99	685.4	.6049	33.61	685.3	.5936	30.79	685.2	.5833	130
140	37.62	690.6	.6136	34.19	690.5	.6023	31.33	690.4	.5920	140
150	38.26	695.8	1.6222	34.77	695.7	1.6109	31.86	695.6	1.6006	150
160	38.90	701.1	.6307	35.35	700.9	.6194	32.39	700.8	.6092	160
170	39.54	706.3	.6391	35.93	706.2	.6278	32.92	706.1	.6176	170
180	40.17	711.6	.6474	36.51	711.5	.6362	33.46	711.4	.6259	180
190	40.81	716.9	.6556	37.09	716.8	.6444	33.99	716.7	.6341	190
200	41.45	722.2	1.6637	37.67	722.1	1.6525	34.52	722.0	1.6422	200

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft.^{3}/lb; H = heat content in Btu./lb; S = entropy in Btu./lb °F.)

Temp. °F.	Absolute pressure in lbs./in.^{2} (Saturation temperature in italics) 8 (*−48.64°*) V	H	S	9 (*−44.38°*) V	H	S	10 (*−41.34°*) V	H	S	Temp. °F.
Sat.	*31.79*	*594.2*	*1.4452*	*28.48*	*595.7*	*1.4363*	*25.81*	*597.1*	*1.4276*	Sat.
−50	…	…	…	…	…	…	…	…	…	−50
−40	32.52	598.8	1.4573	28.85	598.3	1.4426	25.90	597.8	1.4293	−40
−30	33.36	604.1	.4697	29.59	603.6	.4551	26.58	603.2	.4420	−30
−20	34.19	609.3	.4816	30.34	608.9	.4672	27.26	608.5	.4542	−20
−10	35.01	614.4	.4932	31.07	614.0	.4788	27.92	613.7	.4659	−10
0	35.83	619.5	1.5044	31.80	619.2	1.4902	28.58	618.9	1.4773	0
10	36.64	624.6	.5154	32.53	624.3	.5012	29.24	624.0	.4884	10
20	37.45	629.7	.5261	33.26	629.4	.5119	29.90	629.1	.4992	20
30	38.26	634.7	.5365	33.98	634.5	.5224	30.55	634.2	.5097	30
40	39.07	639.8	.5467	34.70	639.5	.5327	31.20	639.3	.5200	40
50	39.88	644.8	1.5568	35.42	644.6	1.5427	31.85	644.4	1.5301	50
60	40.68	649.9	.5666	36.13	649.7	.5526	32.49	649.5	.5400	60
70	41.48	655.0	.5763	36.85	654.8	.5623	33.14	654.6	.5497	70
80	42.28	660.1	.5858	37.56	659.9	.5718	33.78	659.7	.5593	80
90	43.08	665.2	.5952	38.27	665.0	.5812	34.42	664.8	.5687	90
100	43.88	670.3	1.6044	38.98	670.1	1.5904	35.07	670.0	1.5779	100
110	44.68	675.4	.6135	39.70	675.3	.5995	35.71	675.1	.5870	110
120	45.48	680.5	.6224	40.40	680.4	.6085	36.35	680.3	.5960	120
130	46.27	685.7	.6312	41.11	685.6	.6173	36.99	685.4	.6049	130
140	47.07	690.9	.6399	41.82	690.7	.6260	37.62	690.6	.6136	140
150	47.87	696.1	1.6485	42.53	695.9	1.6346	38.26	695.8	1.6222	150
160	48.66	701.3	.6570	43.24	701.2	.6431	38.90	701.1	.6307	160
170	49.46	706.5	.6654	43.95	706.4	.6515	39.54	706.3	.6391	170
180	50.25	711.8	.6737	44.65	711.7	.6598	40.17	711.6	.6474	180

Temp. °F.	13 (*−32.57°*) V	H	S	14 (*−29.76°*) V	H	S	15 (*−27.29°*) V	H	S	Temp. °F.
Sat.	*20.30*	*600.5*	*1.4067*	*18.85*	*601.4*	*1.3996*	*17.67*	*602.4*	*1.3938*	Sat.
−30	20.33	601.8	1.4088	…	…	…	…	…	…	−30
−20	20.86	607.2	.4213	19.33	606.8	1.4119	18.01	606.4	1.4031	−20
−10	21.38	612.6	.4334	19.82	612.2	.4241	18.47	611.9	.4154	−10
0	21.90	617.9	1.4450	20.30	617.6	1.4358	18.92	617.2	1.4272	0
10	22.41	623.1	.4563	20.78	622.8	.4472	19.37	622.5	.4386	10
20	22.92	628.3	.4672	21.26	628.0	.4582	19.82	627.8	.4497	20
30	23.43	633.5	.4779	21.73	633.2	.4688	20.26	633.0	.4604	30
40	23.93	638.6	.4883	22.20	638.4	.4793	20.70	638.2	.4700	40
50	24.43	643.8	1.4985	22.67	643.6	1.4896	21.14	643.4	1.4812	50
60	24.94	648.9	.5085	23.14	648.7	.4996	21.58	648.5	.4912	60
70	25.43	654.1	.5183	23.60	653.9	.5094	22.01	653.7	.5011	70
80	25.93	659.2	.5279	24.06	659.0	.5191	22.44	658.9	.5108	80
90	26.43	664.4	.5374	24.53	664.2	.5285	22.88	664.0	.5203	90
100	26.93	669.5	1.5467	24.99	669.4	1.5378	23.31	669.2	1.5296	100
110	27.42	674.7	.5558	25.45	674.5	.5470	23.74	674.4	.5388	110
120	27.92	679.9	.5649	25.91	679.7	.5560	24.17	679.6	.5478	120
130	28.41	685.1	.5737	26.37	884.9	.5649	24.60	684.8	.5567	130
140	28.90	690.3	.5825	26.83	690.1	.5737	25.03	690.0	.5655	140
150	29.40	695.5	1.5911	27.29	695.4	1.5824	25.46	695.3	1.5742	150
160	29.89	700.7	.5997	27.74	700.6	.5909	25.88	700.5	.5827	160
170	30.38	706.0	.6081	28.20	705.9	.5993	26.31	705.8	.5911	170
180	30.87	711.3	.6164	28.66	711.2	.6076	26.74	711.1	.5995	180
190	31.36	716.6	.6246	29.11	716.5	.6159	27.16	716.4	.6077	190
200	31.85	721.9	1.6328	29.57	721.8	1.6240	27.59	721.7	1.6158	200

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft. ³/lb; H = heat content in Btu. /lb; S = entropy in Btu. /lb °F.)

Temp. °F.	15 (*−27.29°*) V	H	S	16 (*−24.95°*) V	H	S	17 (*−22.75°*) V	H	S	Temp. °F.
Sat.	*17.67*	*602.4*	*1.3938*	*16.64*	*603.2*	*1.3885*	*15.72*	*604.0*	*1.3835*	Sat.
−20	18.01	606.4	1.4031	16.86	606.0	1.3948	15.83	605.6	1.3870	−20
−10	18.47	611.9	.4154	17.29	611.5	.4072	16.24	611.1	.3994	−10
0	18.92	617.2	1.4272	17.72	616.9	1.4191	16.65	616.6	1.4114	0
10	19.37	622.5	.4386	18.14	622.2	.4306	17.05	621.9	.4230	10
20	19.82	627.8	.4497	18.56	627.5	.4417	17.45	627.2	.4342	20
30	20.26	633.0	.4604	18.97	632.7	.4525	17.84	632.5	.4450	30
40	20.70	638.2	.4709	19.39	638.0	.4630	18.23	637.7	.4556	40
50	21.14	643.4	1.4812	19.80	643.2	1.4733	18.62	642.9	1.4659	50
60	21.58	648.5	.4912	20.21	648.3	.4834	19.01	648.1	.4761	60
70	22.01	653.7	.5011	20.62	653.5	.4933	19.39	653.3	.4860	70
80	22.44	658.9	.5108	21.03	658.7	.5030	19.78	658.5	.4957	80
90	22.88	664.0	.5203	21.43	663.9	.5125	20.16	663.7	.5052	90
100	23.31	669.2	1.5296	21.84	669.1	1.5218	20.54	668.9	1.5146	100
110	23.74	674.4	.5388	22.24	674.3	.5310	20.92	674.1	.5238	110
120	24.17	679.6	.5478	22.65	679.5	.5401	21.30	679.3	.5328	120
130	24.60	684.8	.5567	23.05	684.7	.5490	21.68	684.5	.5418	130
140	25.03	690.0	.5655	23.45	689.9	.5578	22.06	689.8	.5506	140
150	25.46	695.3	1.5742	23.86	695.1	1.5665	22.44	695.0	1.5593	150
160	25.88	700.5	.5827	24.26	700.4	.5750	22.82	700.3	.5678	160
170	26.31	705.8	.5911	24.66	705.7	.5835	23.20	705.6	.5763	170
180	26.74	711.1	.5995	25.06	711.0	.5918	23.58	710.9	.5846	180
190	27.16	716.4	.6077	25.46	716.3	.6001	23.95	716.2	.5929	190
200	27.59	721.7	1.6158	25.86	721.6	1.6082	24.33	721.5	1.6010	200
220	28.44	732.4	.6318	26.66	732.3	.6242	25.08	732.2	.6170	220

Temp. °F.	20 (*−16.64°*) V	H	S	21 (*−14.78°*) V	H	S	22 (*−12.98°*) V	H	S	Temp. °F.
Sat.	*13.50*	*606.2*	*1.3700*	*12.90*	*606.8*	*1.3659*	*12.35*	*607.4*	*1.3621*	Sat.
−10	13.74	610.0	1.3784	13.06	609.6	1.3720	12.45	609.2	1.3659	−10
0	14.09	615.5	1.3907	13.40	615.2	1.3844	12.77	614.8	1.3784	0
10	14.44	621.0	.4025	13.73	620.7	.3962	13.09	620.4	.3903	10
20	14.78	626.4	.4138	14.06	626.1	.4077	13.40	625.8	.4018	20
30	15.11	631.7	.4248	14.38	631.5	.4187	13.71	631.2	.4129	30
40	15.45	637.0	.4356	14.70	636.8	.4295	14.02	636.6	.4237	40
50	15.78	642.3	1.4460	15.02	642.1	1.4400	14.32	641.9	1.4342	50
60	16.12	647.5	.4562	15.34	647.3	.4502	14.63	647.1	.4445	60
70	16.45	652.8	.4662	15.65	652.6	.4602	14.93	652.4	.4545	70
80	16.78	658.0	.4760	15.97	657.8	.4700	15.23	657.7	.4643	80
90	17.10	663.2	.4856	16.28	663.1	.4796	15.53	662.9	.4740	90
100	17.43	668.5	1.4950	16.59	668.3	1.4891	15.83	668.1	1.4834	100
110	17.76	673.7	.5042	16.90	673.5	.4983	16.12	673.4	.4927	110
120	18.08	678.9	.5133	17.21	678.8	.5075	16.42	678.6	.5019	120
130	18.41	684.2	.5223	17.52	684.0	.5165	16.72	683.9	.5109	130
140	18.73	689.4	.5312	17.83	689.3	.5253	17.01	689.2	.5197	140
150	19.05	694.7	1.5399	18.14	694.6	1.5340	17.31	694.4	1.5285	150
160	19.37	700.0	.5485	18.44	699.8	.5426	17.60	699.7	.5371	160
170	19.70	705.3	.5569	18.75	705.1	.5510	17.89	705.0	.5456	170
180	20.02	710.6	.5653	19.06	710.5	.5595	18.19	710.4	.5539	180
190	20.34	715.9	.5736	19.36	715.8	.5678	18.48	715.7	.5622	190
200	20.66	721.2	1.5817	19.67	721.1	1.5759	18.77	721.1	1.5704	200
220	21.30	732.0	.5978	20.28	731.9	.5920	19.35	731.8	.5865	220
240	21.94	742.8	.6135	20.89	742.7	.6077	19.94	742.7	.6022	240

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft.^{3}/lb; H = heat content in Btu. /lb; S = entropy in Btu. /lb °F.)

Temp. °F.	18 (−20.81°) V	18 H	18 S	19 (−18.68°) V	19 H	19 S	20 (−16.64°) V	20 H	20 S	Temp. °F
	Absolute pressure in lbs./in.² (Saturation temperature in italics.)									
Sat.	*14.90*	*604.8*	*1.3787*	*14.17*	*605.5*	*1.3748*	*13.50*	*606.2*	*1.3700*	Sat.
−20	14.93	605.1	1.3795	....	....	....	....	....	....	−20
−10	15.32	610.7	.3921	14.49	610.3	1.3851	13.74	610.0	1.3784	−10
0	15.70	616.2	1.4042	14.85	615.9	1.3973	14.09	615.5	1.3907	0
10	16.08	621.6	.4158	15.21	621.3	.4090	14.44	621.0	.4025	10
20	16.46	626.9	.4270	15.57	626.7	.4203	14.78	626.4	.4138	20
30	16.83	632.2	.4380	15.93	632.0	.4312	15.11	631.7	.4248	30
40	17.20	637.5	.4486	16.28	637.3	.4419	15.45	637.0	.4356	40
50	17.57	642.7	1.4590	16.63	642.5	1.4523	15.78	642.3	1.4460	50
60	17.94	647.9	.4691	16.98	647.7	.4625	16.12	647.5	.4562	60
70	18.30	653.1	.4790	17.33	653.0	.4724	16.45	652.8	.4662	70
80	18.67	658.4	.4887	17.67	658.2	.4822	16.78	858.0	.4760	80
90	19.03	663.6	.4983	18.02	663.4	.4918	17.10	663.2	.4856	90
100	19.39	668.8	1.5077	18.36	668.6	1.5012	17.43	668.5	1.4950	100
110	19.75	674.0	.5169	18.70	673.8	.5104	17.76	673.7	.5042	110
120	20.11	679.2	.5260	19.04	679.1	.5195	18.08	678.9	.5133	120
130	20.47	684.4	.5349	19.38	684.3	.5285	18.41	684.2	.5223	130
140	20.83	689.7	.5438	19.72	689.5	.5373	18.73	689.4	.5312	140
150	21.19	694.9	1.5525	20.06	694.8	1.5460	19.05	694.7	1.5399	150
160	21.54	700.2	.5610	20.40	700.1	.5546	19.37	700.0	.5485	160
170	21.90	705.5	.5695	20.74	705.4	.5631	19.70	705.3	.5569	170
180	22.26	710.8	.5778	21.08	710.7	.5714	20.02	710.6	.5653	180
190	22.61	716.1	.5861	21.42	716.0	.5797	20.34	715.9	.5736	190
200	22.97	721.4	1.5943	21.75	721.3	1.5878	20.66	721.2	1.5817	200
220	23.68	732.2	.6103	22.43	732.1	.6039	21.30	732.0	.5978	220

Temp. °F.	23 (−11.66°) V	23 H	23 S	24 (−9.68°) V	24 H	24 S	25 (−7.96°) V	25 H	25 S	Temp. °F
Sat.	*11.88*	*608.1*	*1.3684*	*11.39*	*608.8*	*1.3649*	*10.96*	*609.1*	*1.3618*	Sat.
−10	11.89	608.8	1.3600	....	....	....	....	....	....	−10
0	12.20	614.5	1.3728	11.67	614.1	1.3670	11.19	613.8	1.3616	0
10	12.50	620.0	.3846	11.96	619.7	.3791	11.47	619.4	.3738	10
20	12.80	625.5	.3961	12.25	625.2	.3907	11.75	625.0	.3855	20
30	13.10	630.9	.4073	12.54	630.7	.4019	12.03	630.4	.3967	30
40	13.40	636.3	.4181	12.82	636.1	.4128	12.30	635.8	.4077	40
50	13.69	641.6	1.4287	13.11	641.4	1.4234	12.57	641.2	1.4183	50
60	13.98	646.9	.4390	13.39	646.7	.4337	12.84	646.5	.4287	60
70	14.27	652.2	.4491	13.66	652.0	.4438	13.11	651.8	.4388	70
80	14.56	657.5	.4589	13.94	657.3	.4537	13.37	657.1	.4487	80
90	14.84	662.7	.4686	14.22	662.6	.4634	13.64	662.4	.4584	90
100	15.13	668.0	1.4780	14.49	667.8	1.4729	13.90	667.7	1.4679	100
110	15.41	673.2	.4873	14.76	673.1	.4822	14.17	673.0	.4772	110
120	15.70	678.5	.4965	15.04	678.4	.4914	14.43	678.2	.4864	120
130	15.98	683.8	.5055	15.31	683.6	.5004	14.69	683.5	.4954	130
140	16.26	689.0	.5144	15.58	688.9	.5093	14.95	688.8	.5043	140
150	16.55	694.3	1.5231	15.85	694.2	1.5180	15.21	694.1	1,5131	150
160	16.83	699.6	.5317	16.12	699.5	.5266	15.47	699.4	.5217	160
170	17.11	704.9	.5402	16.39	704.8	.5352	15.73	704.7	.5303	170
180	17.39	710.3	.5486	16.66	710.2	.5436	15.99	710.1	.5387	180
190	17.67	715.6	.5569	16.93	715.5	.5518	16.25	715.4	.5470	190
200	17.95	721.0	1.5651	17.20	720.9	1.5600	16.50	720.8	1.5552	200
220	18.51	731.7	.5812	17.73	731.7	.5761	17.02	731.6	.5713	220
240	19.07	742.6	.5969	18.27	742.6	.5919	17.53	742.5	.5870	240

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft.^{3}/lb; H = heat content in Btu. /lb; S = entropy in Btu. /lb °F.)

Temp. °F.	Absolute pressure in lbs. / in.^{2} (Saturation temperature in italics.)									Temp. °F.
	25 (*−7.96°*)			26 (*−6.39°*)			27 (*−4.87°*)			
	V	H	S	V	H	S	V	H	S	
Sat.	*10.96*	*609.1*	*1.3516*	*10.56*	*609.7*	*1.3458*	*10.20*	*610.2*	*1.3451*	Sat.
0	11.19	613.8	1.3616	10.74	613.4	1.3564	10.33	613.0	1.3513	0
10	11.47	619.4	.3738	11.01	619.1	.3686	10.59	618.8	.3637	10
20	11.75	625.0	.3855	11.28	624.7	.3804	10.85	624.4	.3755	20
30	12.03	630.4	.3967	11.55	630.2	.3917	11.11	629.9	.3869	30
40	12.30	635.8	.4077	11.81	635.6	.4027	11.37	635.4	.3979	40
50	12.57	641.2	1.4183	12.08	641.0	1.4134	11.62	640.8	1.4087	50
60	12.84	646.5	.4287	12.34	646.3	.4238	11.87	646.1	.4191	60
70	13.11	651.8	.4388	12.59	651.6	.4339	12.12	651.5	.4292	70
80	13.37	657.1	.4487	12.85	656.9	.4439	12.37	656.8	.4392	80
90	13.64	662.4	.4584	13.11	662.2	.4536	12.61	662.1	.4489	90
100	13.90	667.7	1.4679	13.36	667.5	1.4631	12.86	667.4	1.4585	100
110	14.17	673.0	.4772	13.61	672.8	.4725	13.10	672.7	.4679	110
120	14.43	678.2	.4864	13.87	678.1	.4817	13.34	678.0	.4771	120
130	14.69	683.5	.4954	14.12	683.4	.4907	13.59	683.3	.4861	130
140	14.95	688.8	.5043	14.37	688.7	.4996	13.83	688.6	.4950	140
150	15.21	694.1	1.5131	14.62	694.0	1.5084	14.07	693.9	1.5038	150
160	15.47	699.4	.5217	14.87	699.3	.5170	14.31	699.2	.5125	160
170	15.73	704.7	.5303	15.12	704.6	.5256	14.55	704.5	.5210	170
180	15.99	710.1	.5387	15.37	710.0	.5340	14.79	709.9	.5295	180
190	16.25	715.4	.5470	15.62	715.3	.5423	15.03	715.2	.5378	190
200	16.50	720.8	1.5552	15.86	720.7	1.5505	15.27	720.6	1.5460	200
220	17.02	731.6	.5713	16.36	731.5	.5666	15.75	731.4	.5621	220
240	17.53	742.5	.5870	16.85	742.4	.5824	16.23	742.3	.5779	240
260	18.04	753.4	.6025	17.35	753.3	.5978	16.70	753.2	.5933	260

Temp. °F.	30 (*−0.57°*)			31 (*+0.79°*)			32 (*+2.11°*)			Temp. °F.
Sat.	*9.236*	*611.6*	*1.3364*	*8.965*	*612.0*	*1.3336*	*8.695*	*612.4*	*1.3310*	Sat.
10	9.492	617.8	1.3497	9.173	617.4	1.3453	8.874	617.1	1.3411	10
20	9.731	623.5	.3618	9.405	623.2	.3574	9.099	622.9	.3532	20
30	9.966	629.1	.3733	9.633	628.8	.3691	9.321	628.5	.3649	30
40	10.20	634.6	.3845	9.858	634.4	.3803	9.540	634.1	.3762	40
50	10.43	640.1	1.3953	10.08	639.9	1.3912	9.757	639.6	1.3871	50
60	10.65	645.5	.4059	10.30	645.3	.4017	9.972	645.1	.3977	60
70	10.88	650.9	.4161	10.52	650.7	.4120	10.18	650.5	.4080	70
80	11.10	656.2	.4261	10.74	656.1	.4221	10.40	655.9	.4181	80
90	11.33	661.6	.4359	10.96	661.4	.4319	10.61	661.2	.4280	90
100	11.55	666.9	1.4456	11.17	666.7	1.4415	10.81	666.6	1.4376	100
110	11.77	672.2	.4550	11.38	672.1	.4510	11.02	671.9	.4470	110
120	11.99	677.5	.4642	11.60	677.4	.4602	11.23	677.3	.4563	120
130	12.21	682.9	.4733	11.81	682.7	.4693	11.44	682.6	.4655	130
140	12.43	688.2	.4823	12.02	688.1	.4783	11.64	687.9	.4744	140
150	12.65	693.5	1.4911	12.23	693.4	1.4871	11.85	693.3	1.4833	150
160	12.87	698.8	.4998	12.44	698.7	.4958	12.05	698.6	.4920	160
170	13.08	704.2	.5083	12.66	704.1	.5044	12.26	704.0	.5006	170
180	13.30	709.6	.5168	12.87	709.5	.5129	12.46	709.4	.5090	180
190	13.52	714.9	.5251	13.07	714.8	.5212	12.66	714.7	.5174	190
200	13.73	720.3	1.5334	13.28	720.2	1.5294	12.86	720.1	1.5256	200
220	14.16	731.1	.5495	13.70	731.1	.5456	13.27	731.0	.5418	220
240	14.59	742.0	.5653	14.12	742.0	.5614	13.67	741.9	.5576	240
260	15.02	753.0	.5808	14.53	752.9	.5769	14.08	752.9	.5731	260
280	15.45	764.1	.5960	14.95	764.0	.5921	14.48	763.9	.5883	280

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft.^{3}/lb; H = heat content in Btu. /lb; S = entropy in Btu. /lb °F.)

Temp. °F.	Absolute pressure in lbs./in.^{2} (Saturation temperature in italics)									Temp. °F.
	28 *−3.40°*			29 *−1.97°*			30 *−0.57°*			
	V	H	S	V	H	S	V	H	S	
Sat.	*9.853*	*610.7*	*1.3481*	*9.534*	*611.1*	*1.3422*	*9.236*	*611.6*	*1.3364*	Sat.
0	9.942	612.7	1.3465	9.584	612.3	1.3417	9.250	611.9	1.3371	0
10	10.20	618.4	.3589	9.834	618.1	.3542	9.492	617.8	.3497	10
20	10.45	624.1	.3708	10.08	623.8	.3662	9.731	623.5	.3618	20
30	10.70	629.6	.3822	10.32	629.4	.3777	9.966	629.1	.3733	30
40	10.95	635.1	.3933	10.56	634.9	.3888	10.20	634.6	.3845	40
50	11.19	640.5	1.4041	10.80	640.3	1.3996	10.43	640.1	1.3953	50
60	11.44	645.9	.4145	11.03	645.7	.4101	10.65	645.5	.4059	60
70	11.68	651.2	.4247	11.26	651.1	.4204	10.88	650.9	.4161	70
80	11.92	656.6	.4347	11.50	656.4	.4304	11.10	656.2	.4261	80
90	12.15	661.9	.4445	11.73	661.7	.4401	11.33	661.6	.4359	90
100	12.39	667.2	1.4540	11.96	667.1	1.4497	11.55	666.9	1.4456	100
110	12.63	672.5	.4634	12.18	672.4	.4591	11.77	672.2	.4550	110
120	12.86	677.8	.4726	12.41	677.7	.4684	11.99	677.5	.4642	120
130	13.10	683.1	.4817	12.64	683.0	.4775	12.21	682.9	.4733	130
140	13.33	688.4	.4906	12.86	688.3	.4864	12.43	688.2	.4823	140
150	13.56	693.7	1.4994	13.09	693.6	1.4952	12.65	693.5	1.4911	150
160	13.80	699.1	.5081	13.31	699.0	.5039	12.87	698.8	.4998	160
170	14.03	704.4	.5167	13.54	704.3	.5124	13.08	704.2	.5083	170
180	14.26	709.8	.5251	13.76	709.7	.5209	13.30	709.6	.5168	180
190	14.49	715.1	.5334	13.99	715.0	.5292	13.52	714.9	.5251	190
200	14.72	720.5	1.5416	14.21	720.4	.5374	13.73	720.3	1.5334	200
220	15.18	731.3	.5578	14.65	731.2	.5536	14.16	731.1	.5495	220
240	15.64	742.2	.5736	15.10	742.2	.5694	14.59	742.0	.5653	240
260	16.10	753.2	.5890	15.54	753.1	.5848	15.02	753.0	.5808	260
	33 *3.40°*			34 *4.66°*			35 *5.89°*			
Sat.	*8.445*	*612.8*	*1.3285*	*8.211*	*613.2*	*1.3260*	*7.991*	*613.6*	*1.3236*	Sat.
10	8.592	616.8	1.3369	8.328	616.4	1.3328	8.078	616.1	1.3289	10
20	8.812	622.6	.3492	8.542	622.3	.3452	8.287	622.0	.3413	20
30	9.028	628.3	.3609	8.753	628.0	.3570	8.493	627.7	.3532	30
40	9.242	633.9	.3722	8.960	633.6	.3684	8.695	633.4	.3646	40
50	9.452	639.4	1.3832	9.166	639.2	1.3793	8.895	638.9	1.3756	50
60	9.661	644.9	.3938	9.369	644.7	.3900	9.093	644.4	.3863	60
70	9.868	650.3	.4042	9.570	650.1	.4004	9.289	649.9	.3967	70
80	10.07	655.7	.4143	9.770	655.5	.4105	9.484	655.3	.4069	80
90	10.28	661.1	.4241	9.969	660.9	.4204	9.677	660.7	.4168	90
100	10.48	666.4	1.4338	10.17	666.3	1.4301	9.869	666.1	1.4265	100
110	10.68	671.8	.4433	10.36	671.6	.4396	10.06	671.5	.4360	110
120	10.88	677.1	.4526	10.56	677.0	.4489	10.25	676.8	.4453	120
130	11.08	682.5	.4617	10.75	682.3	.4581	10.44	682.2	.4545	130
140	11.28	687.8	.4707	10.95	687.7	.4671	10.63	687.6	.4635	140
150	11.48	693.2	1.4795	11.14	693.0	1.4759	10.82	692.9	1.4724	150
160	11.68	698.5	.4883	11.33	698.4	.4846	11.00	698.3	.4811	160
170	11.88	703.9	.4968	11.53	703.8	.4932	11.19	703.7	.4897	170
180	12.08	709.3	.5053	11.72	709.2	.5017	11.38	709.1	.4982	180
190	12.27	714.6	.5137	11.91	714.5	.5101	11.56	714.5	.5066	190
200	12.47	720.0	1.5219	12.10	720.0	1.5183	11.75	719.9	1.5148	200
220	12.86	730.9	.5381	12.48	730.8	.5346	12.12	730.7	.5311	220
240	13.26	741.8	.5540	12.86	741.7	.5504	12.49	741.7	.5469	240
260	13.65	752.8	.5695	13.24	752.7	.5659	12.86	752.7	.5624	260
280	14.04	763.9	.5846	13.62	763.8	.5811	13.23	763.7	.5776	280

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft.^{3}/lb; H = heat content in Btu. /lb; S = entropy in Btu. /lb °F.)

Absolute pressure in lbs./in.^{2} (Saturation temperature in italics.)

Temp. °F.	35 (*5.89°*) V	H	S	36 (*7.09°*) V	H	S	37 (*8.27°*) V	H	S	Temp. °F.
Sat.	*7.991*	*613.6*	*1.3236*	*7.782*	*614.0*	*1.3213*	*7.584*	*614.3*	*1.3190*	Sat.
10	8.078	616.1	1.3289	7.842	615.7	1.3250	7.619	615.4	1.3212	10
20	8.287	622.0	.3413	8.046	621.7	.3376	7.819	621.4	.3338	20
30	8.493	627.7	.3532	8.247	627.4	.3494	8.015	627.2	.3458	30
40	8.695	633.4	.3646	8.445	633.1	.3609	8.208	632.9	.3573	40
50	8.895	638.9	1.3756	8.640	638.7	1.3720	8.398	638.5	1.3684	50
60	9.093	644.4	.3863	8.833	644.2	.3827	8.587	644.0	.3792	60
70	9.289	649.9	.3967	9.024	649.7	.3932	8.773	649.5	.3897	70
80	9.484	655.3	.4069	9.214	655.2	.4033	8.958	655.0	.3999	80
90	9.677	660.7	.4168	9.402	660.6	.4133	9.142	660.4	.4098	90
100	9.869	666.1	1.4265	9.589	666.0	1.4230	9.324	665.8	1.4196	100
110	10.06	671.5	.4360	9.775	671.3	.4325	9.506	671.2	.4291	110
120	10.25	676.8	.4453	9.961	676.7	.4419	9.686	676.6	.4385	120
130	10.44	682.2	.4545	10.15	682.1	.4510	9.866	681.9	.4477	130
140	10.63	687.6	.4635	10.33	687.4	.4601	10.05	687.3	.4567	140
150	10.82	692.9	1.4724	10.51	692.8	1.4689	10.22	692.7	1.4656	150
160	11.00	698.3	.4811	10.69	698.2	.4777	10.40	698.1	.4744	160
170	11.19	703.7	.4897	10.88	703.6	.4863	10.58	703.5	.4830	170
180	11.38	709.1	.4982	11.06	709.0	.4948	10.76	708.9	.4915	180
190	11.56	714.5	.5066	11.24	714.4	.5032	10.93	714.3	.4999	190
200	11.75	719.9	1.5148	11.42	719.8	1.5115	11.11	719.7	1.5082	200
220	12.12	730.7	.5311	11.78	730.6	.5277	11.46	730.6	.5244	220
240	12.49	741.7	.5469	12.14	741.6	.5436	11.81	741.5	.5403	240
260	12.86	752.7	.5624	12.50	752.6	.5591	12.16	752.5	.5558	260
280	13.23	763.7	.5776	12.86	763.7	.5743	12.51	763.6	.5710	280

Temp. °F.	40 (*11.66°*) V	H	S	42 (*13.81°*) V	H	S	44 (*15.85°*) V	H	S	Temp. °F.
Sat.	*7.047*	*615.4*	*1.3125*	*6.731*	*616.0*	*1.3084*	*6.442*	*616.6*	*1.3046*	Sat.
20	7.203	620.4	1.3231	6.842	619.8	1.3164	6.513	619.1	1.3099	20
30	7.387	626.3	.3353	7.019	625.8	.3287	6.683	625.2	.3224	30
40	7.568	632.1	.3470	7.192	631.6	.3405	6.850	631.1	.3343	40
50	7.746	637.8	1.3583	7.363	637.3	1.3519	7.014	636.8	1.3457	50
60	7.922	643.4	.3692	7.531	643.0	.3628	7.176	642.5	.3567	60
70	8.096	648.9	.3797	7.697	648.5	.3734	7.336	648.1	.3674	70
80	8.268	654.4	.3900	7.862	654.1	.3838	7.494	653.7	.3778	80
90	8.439	659.9	.4000	8.026	659.5	.3939	7.650	659.2	.3880	90
100	8.609	665.3	1.4098	8.188	665.0	1.4037	7.806	664.7	1.3978	100
110	8.777	670.7	.4194	8.349	670.4	.4133	7.960	670.1	.4075	110
120	8.945	676.1	.4288	8.510	675.9	.4228	8.114	675.6	.4170	120
130	9.112	681.5	.4381	8.669	681.3	.4320	8.267	681.0	.4263	130
140	9.278	686.9	.4471	8.828	686.7	.4411	8.419	686.4	.4354	140
150	9.444	692.3	1.4561	8.986	692.1	1.4501	8.570	691.9	1.4444	150
160	9.609	697.7	.4648	9.144	697.5	.4589	8.721	697.3	.4532	160
170	9.774	703.1	.4735	9.301	702.9	.4676	8.871	702.7	.4619	170
180	9.938	708.5	.4820	9.458	708.3	.4761	9.021	708.1	.4704	180
190	10.10	714.0	.4904	9.614	713.8	.4845	9.171	713.6	.4789	190
200	10.27	719.4	1.4987	9.770	719.2	1.4928	9.320	719.0	1.4872	200
220	10.59	730.3	.5150	10.08	730.1	.5091	9.617	730.0	.5035	220
240	10.92	741.3	.5309	10.39	741.1	.5251	9.913	741.0	.5195	240
260	11.24	752.3	.5465	10.70	752.2	.5406	10.21	752.0	.5350	260
280	11.56	763.4	.5617	11.01	763.3	.5559	10.50	763.1	.5503	280
300	11.88	774.6	.5766	11.31	774.5	1.5708	10.80	774.3	1.5652	300

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft.^{3}/lb; H = heat content in Btu. /lb; S = entropy in Btu. /lb °F.)

Temp. °F.	Absolute pressure in lbs./in.² (Saturation temperature in italics.)									Temp. °F.
	38 *9.42°*			39 *10.55°*			40 *11.68°*			
	V	H	S	V	H	S	V	H	S	
Sat.	*7.396*	*614.7*	*1.3168*	*7.217*	*615.0*	*1.3146*	*7.047*	*615.4*	*1.3125*	Sat.
10	7.407	615.0	1.3175	.....	.....	.....	.....	.....	.....	10
20	7.603	621.0	.3301	7.398	620.7	1.3266	7.203	620.4	1.3231	20
30	7.795	626.9	.3422	7.586	626.6	.3387	7.387	626.3	.3353	30
40	7.983	632.6	.3538	7.770	632.4	.3504	7.568	632.1	.3470	40
50	8.170	638.3	1.3650	7.952	638.0	1.3616	7.746	637.8	1.3583	50
60	8.353	643.8	.3758	8.132	643.6	.3724	7.922	643.4	.3692	60
70	8.535	649.3	.3863	8.310	649.1	.3830	8.096	648.9	.3797	70
80	8.716	654.8	.3965	8.486	654.6	.3932	8.268	654.4	.3900	80
90	8.895	660.2	.4065	8.661	660.1	.4032	8.439	659.9	.4000	90
100	9.073	665.6	1.4163	8.835	665.5	1.4130	8.609	665.3	1.4098	100
110	9.250	671.0	.4258	9.008	670.9	.4226	8.777	670.7	.4194	110
120	9.426	676.4	.4352	9.179	676.3	.4320	8.945	676.1	.4288	120
130	9.602	681.8	.4444	9.351	681.7	.4412	9.112	681.5	.4381	130
140	9.776	687.2	.4534	9.521	687.1	.4503	9.278	686.9	.4471	140
150	9.950	692.6	1.4623	9.691	692.5	1.4592	9.444	692.3	1.4561	150
160	10.12	698.0	.4711	9.860	697.8	.4679	9.609	697.7	.4648	160
170	10.30	703.3	.4797	10.03	703.2	.4766	9.774	703.1	.4735	170
180	10.47	708.7	.4883	10.20	708.6	.4851	9.938	708.5	.4820	180
190	10.64	714.2	.4966	10.36	714.1	.4935	10.10	714.0	.4904	190
200	10.81	719.6	1.5049	10.53	719.5	1.5018	10.27	719.4	1.4987	200
220	11.16	730.5	.5212	10.87	730.4	.5181	10.59	730.3	.5150	220
240	11.50	741.4	.5371	11.20	741.3	.5340	10.92	741.3	.5309	240
260	11.84	752.4	.5526	11.53	752.4	.5495	11.24	752.3	.5465	260
280	12.18	763.5	.5678	11.86	763.5	.5647	11.56	763.4	.5617	280
	46 *17.87°*			48 *19.80°*			50 *21.67°*			
Sat.	*6.177*	*617.2*	*1.3009*	*5.934*	*617.7*	*1.2973*	*5.710*	*618.2*	*1.2939*	Sat.
20	6.213	618.5	1.3036	5.937	617.8	1.2976	.....	.....	.....	20
30	6.377	624.6	.3102	6.096	624.0	.3103	5.838	623.4	1.3046	30
40	6.538	630.5	.3283	6.251	630.0	.3225	5.988	629.5	.3169	40
50	6.696	636.4	1.3398	6.404	635.9	1.3341	6.135	635.4	1.3286	50
60	6.851	642.1	.3509	6.554	641.6	.3453	6.280	641.2	.3399	60
70	7.005	647.7	.3617	6.702	647.3	.3561	6.423	646.9	.3508	70
80	7.157	653.3	.3721	6.848	652.9	.3666	6.564	652.6	.3613	80
90	7.308	658.9	.3823	6.993	658.5	.3768	6.704	658.2	.3716	90
100	7.457	664.4	1.3922	7.137	664.0	1.3868	6.843	663.7	1.3816	100
110	7.605	669.8	.4019	7.280	669.5	.3965	6.980	669.2	.3914	110
120	7.753	675.3	.4114	7.421	675.0	.4061	7.117	674.7	.4009	120
130	7.899	680.7	.4207	7.562	680.5	.4154	7.252	680.2	.4103	130
140	8.045	686.2	.4299	7.702	685.9	.4246	7.387	685.7	.4195	140
150	8.190	691.6	1.4389	7.842	691.4	1.4336	7.521	691.1	1.4286	150
160	8.335	697.1	.4477	7.981	696.8	.4425	7.655	696.6	.4374	160
170	8.479	702.5	.4564	8.119	702.3	.4512	7.788	702.1	.4462	170
180	8.623	707.9	.4650	8.257	707.7	.4598	7.921	707.5	.4548	180
190	8.766	713.4	.4735	8.395	713.2	.4683	8.053	713.0	.4633	190
200	8.909	718.8	1.4818	8.532	718.7	1.4766	8.185	718.5	1.4716	200
220	9.194	729.8	.4981	8.805	729.6	.4930	8.448	729.4	.4880	220
240	9.477	740.8	.5141	9.077	740.6	.5090	8.710	740.5	.5040	240
260	9.760	751.9	.5297	9.348	751.7	.5246	8.970	751.6	.5197	260
280	10.04	763.0	.5450	9.619	762.9	.5399	9.230	762.7	.5350	280
300	10.32	774.2	1.5599	9.888	774.1	1.5548	9.489	774.0	1.5500	300

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft.^{3}/lb; H = heat content in Btu./lb; S = entropy in Btu./lb °F.)

Temp °F	Absolute pressure in lb./in.² (Saturation temperatures in italics.)									Temp °F
	50 *21.67°*			52 *23.45°*			54 *25.23°*			
	V	H	S	V	H	S	V	H	S	
Sat.	*5.710*	*618.2*	*1.2939*	*5.502*	*618.7*	*1.2906*	*5.309*	*619.2*	*1.2875*	*Sat.*
30	5.838	623.4	1.3046	5.599	622.8	1.2991	5.378	622.2	1.2937	30
40	5.988	629.5	.3169	5.744	629.0	.3114	5.519	628.4	.3062	40
50	6.135	635.4	1.3286	5.887	634.9	1.3233	5.657	634.4	1.3181	50
60	6.280	641.2	.3399	6.027	640.8	.3346	5.793	640.3	.3295	60
70	6.423	646.9	.3508	6.165	646.5	.3456	5.927	646.1	.3406	70
80	6.564	652.6	.3613	6.302	652.2	.3562	6.059	651.8	.3513	80
90	6.704	658.2	.3716	6.437	657.8	.3665	6.190	657.5	.3616	90
100	6.843	663.7	1.3816	6.571	663.4	1.3766	6.319	663.1	1.3717	100
110	6.980	669.2	.3914	6.704	668.9	.3864	6.447	668.6	.3816	110
120	7.117	674.7	.4009	6.835	674.4	.3960	6.575	674.2	.3912	120
130	7.252	680.2	.4103	6.966	679.9	.4054	6.701	679.7	.4006	130
140	7.387	685.7	.4195	7.096	685.4	.4146	6.827	685.2	.4099	140
150	7.521	691.1	1.4286	7.225	690.9	1.4237	6.952	690.7	1.4190	150
160	7.655	696.6	.4374	7.354	696.4	.4326	7.076	696.1	.4279	160
170	7.788	702.1	.4462	7.483	701.8	.4413	7.200	701.6	.4367	170
180	7.921	707.5	.4548	7.611	707.3	.4500	7.323	707.1	.4453	180
190	8.053	713.0	.4633	7.738	712.8	.4585	7.446	712.6	.4538	190
200	8.185	718.5	1.4716	7.865	718.3	1.4668	7.569	718.1	1.4622	200
210	8.317	724.0	.4799	7.992	723.8	.4751	7.691	723.6	.4705	210
220	8.448	729.4	.4880	8.118	729.3	.4833	7.813	729.1	.4787	220
240	8.710	740.5	.5040	8.370	740.3	.4993	8.056	740.2	.4947	240
260	8.970	751.6	.5197	8.621	751.4	.5149	8.298	751.3	.5104	260
280	9.230	762.7	1.5350	8.871	762.6	1.5303	8.539	762.5	1.5257	280
300	9.489	774.0	.5500	9.120	773.8	.5453	8.779	773.7	.5407	300
	60 *30.21°*			62 *31.78°*			64 *33.31°*			
Sat.	*4.806*	*620.5*	*1.2787*	*4.668*	*620.9*	*1.2769*	*4.519*	*621.3*	*1.2733*	*Sat.*
40	4.933	626.8	1.2913	4.762	626.2	1.2866	4.602	625.6	1.2820	40
50	5.060	632.9	1.3035	4.886	632.4	1.2989	4.723	631.9	1.2944	50
60	5.184	639.0	.3152	5.007	638.5	.3107	4.842	638.0	.3063	60
70	5.307	644.9	.3265	5.127	644.4	.3220	4.958	644.0	.3177	70
80	5.428	650.7	.3373	5.244	650.3	.3330	5.072	649.9	.3287	80
90	5.547	656.4	.3479	5.360	656.0	.3435	5.185	655.7	.3393	90
100	5.665	662.1	1.3581	5.474	661.7	1.3538	5.296	661.4	1.3496	100
110	5.781	667.7	.3681	5.588	667.4	.3638	5.406	667.1	.3597	110
120	5.897	673.3	.3778	5.700	673.0	.3736	5.516	672.7	.3695	120
130	6.012	678.9	.3873	5.811	678.6	.3831	5.624	678.3	.3791	130
140	6.126	684.4	.3966	5.922	684.2	.3925	5.731	683.9	.3885	140
150	6.239	689.9	1.4058	6.032	689.7	1.4017	5.838	689.5	1.3977	150
160	6.352	695.5	.4148	6.142	695.2	.4107	5.944	695.0	.4067	160
170	6.464	701.0	.4236	6.250	700.8	.4195	6.050	700.5	.4156	170
180	6.576	706.5	.4323	6.359	706.3	.4282	6.155	706.1	.4243	180
190	6.687	712.0	.4409	6.467	711.8	.4368	6.260	711.6	.4329	190
200	6.798	717.5	1.4493	6.574	717.3	1.4453	6.364	717.2	1.4413	200
210	6.909	723.1	.4576	6.681	722.9	.4536	6.468	722.7	.4497	210
220	7.019	728.6	.4658	6.788	728.4	.4618	6.572	728.3	.4579	220
230	7.129	734.1	.4739	6.895	734.0	.4699	6.675	733.8	.4660	230
240	7.238	739.7	.4819	7.001	739.6	.4779	6.778	739.4	.4741	240
260	7.457	750.9	1.4976	7.213	750.7	1.4937	6.984	750.6	1.4898	260
280	7.675	762.1	.5130	7.424	761.9	.5091	7.188	761.8	.5052	280
300	7.892	773.3	.5281	7.634	773.2	.5241	7.392	773.1	.5203	300

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft.³/lb; H = heat content in Btu./lb; S = entropy in Btu./lb °F.)

Temp. °F.	Absolute pressure in lbs./in.² (Saturation temperature in italics.)									Temp. °F.
	56 *26.94°*			58 *28.59°*			60 *30.21°*			
	V	H	S	V	H	S	V	H	S	
Sat.	*5.189*	*619.7*	*1.2844*	*4.962*	*620.1*	*1.2815*	*4.805*	*620.5*	*1.2767*	Sat.
30	5.172	621.6	1.2884	4.981	621.0	1.2834	....	....	....	30
40	5.310	627.9	.3011	5.115	627.3	.2961	4.939	626.8	1.2913	40
50	5.444	633.9	1.3131	5.245	633.4	1.3082	5.060	632.9	1.3035	50
60	5.576	639.9	.3246	5.373	639.4	.3199	5.184	639.0	.3152	60
70	5.706	645.7	.3357	5.499	645.3	.3310	5.307	644.9	.3265	70
80	5.834	651.4	.3465	5.624	651.1	.3418	5.428	650.7	.3373	80
90	5.960	657.1	.3569	5.746	656.8	.3523	5.547	656.4	.3479	90
100	6.085	662.7	1.3670	5.868	662.4	1.3625	5.665	662.1	1.3581	100
110	6.209	668.3	.3769	5.988	668.0	.3724	5.781	667.7	.3681	110
120	6.333	673.9	.3866	6.107	673.6	.3821	5.897	673.3	.3778	120
130	6.455	679.4	.3961	6.226	679.1	.3916	6.012	678.9	.3873	130
140	6.576	684.9	.4053	6.343	684.7	.4009	6.126	684.4	.3966	140
150	6.697	690.4	1.4144	6.460	690.2	1.4100	6.239	689.9	1.4058	150
160	6.817	695.9	.4234	6.577	695.7	.4190	6.352	695.5	.4148	160
170	6.937	701.4	.4322	6.692	701.2	.4278	6.464	701.0	.4236	170
180	7.056	706.9	.4408	6.808	706.7	.4365	6.576	706.5	.4323	180
190	7.175	712.4	.4494	6.923	712.2	.4450	6.687	712.0	.4409	190
200	7.294	717.9	1.4578	7.037	717.7	1.4535	6.798	717.5	1.4493	200
210	7.412	723.4	.4661	7.151	723.2	.4618	6.909	723.1	.4576	210
220	7.529	728.9	.4743	7.265	728.8	.4700	7.019	728.6	.4658	220
240	7.764	740.0	.4903	7.492	739.9	.4860	7.238	739.7	.4819	240
260	7.998	751.1	.5060	7.718	751.0	.5017	7.457	750.9	.4976	260
280	8.230	762.3	1.5213	7.943	762.2	.5171	7.675	762.1	1.5130	280
300	8.462	773.6	.5364	8.167	773.5	.5321	7.892	773.3	.5281	300
	66 *34.81°*			68 *36.27°*			70 *37.70°*			
Sat.	*4.389*	*621.7*	*1.2707*	*4.267*	*622.0*	*1.2682*	*4.151*	*622.4*	*1.2658*	Sat.
40	4.452	625.1	1.2775	4.310	624.5	1.2731	4.177	623.9	1.2688	40
50	4.570	631.4	1.2900	4.426	630.9	1.2858	4.290	630.4	1.2816	50
60	4.686	637.6	.3020	4.539	637.1	.2978	4.401	636.6	.2937	60
70	4.799	643.6	.3135	4.650	643.2	.3094	4.509	642.7	.3054	70
80	4.910	649.5	.3245	4.758	649.1	.3205	4.615	648.7	.3166	80
90	5.020	655.3	.3352	4.865	655.0	.3312	4.719	654.6	.3274	90
100	5.129	661.1	1.3456	4.971	660.7	1.3417	4.822	660.4	1.3378	100
110	5.236	666.8	.3557	5.075	666.5	.3518	4.924	666.1	.3480	110
120	5.342	672.4	.3655	5.179	672.1	.3617	5.025	671.8	.3579	120
130	5.447	678.0	.3751	5.281	677.8	.3713	5.125	677.5	.3676	130
140	5.552	683.6	.3846	5.383	683.4	.3807	5.224	683.1	.3770	140
150	5.656	689.2	1.3938	5.484	689.0	1.3900	5.323	688.7	1.3863	150
160	5.759	694.8	.4028	5.585	694.5	.3991	5.420	694.3	.3954	160
170	5.862	700.3	.4117	5.685	700.1	.4080	5.518	699.9	.4043	170
180	5.964	705.9	.4205	5.784	705.7	.4167	5.615	705.5	.4131	180
190	6.066	711.4	.4291	5.883	711.2	.4254	5.711	711.0	.4217	190
200	6.167	717.0	1.4375	5.982	716.8	1.4338	5.807	716.6	1.4302	200
210	6.268	722.5	.4459	6.080	722.3	.4422	5.902	722.2	.4386	210
220	6.369	728.1	.4541	6.179	727.9	.4505	5.998	727.7	.4469	220
230	6.470	733.7	.4623	6.275	733.5	.4586	6.093	733.3	.4550	230
240	6.570	739.2	.4703	6.373	739.1	.4666	6.187	738.9	.4631	240
260	6.769	750.4	1.4861	6.567	750.3	1.4824	6.376	750.1	1.4789	260
280	6.968	761.7	.5015	6.760	761.5	.4979	6.563	761.4	.4943	280
300	7.165	773.0	.5166	6.952	772.8	.5130	6.750	772.7	.5095	300

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft.³/lb; H = heat content in Btu. /lb; S = entropy in Btu. /lb °F.)

Temp. °F.	75 (41.15°) V	75 H	75 S	80 (44.40°) V	80 H	80 S	85 (47.60°) V	85 H	85 S	Temp. °F.
Sat.	*3.887*	*623.2*	*1.2599*	*3.655*	*624.0*	*1.2545*	*3.449*	*624.7*	*1.2494*	Sat.
50	3.982	629.1	1.2715	3.712	627.7	1.2619	3.473	626.4	1.2527	50
60	4.087	635.5	2839	3.812	634.3	.2745	3.569	633.0	.2656	60
70	4.189	641.7	2957	3.909	640.6	.2866	3.662	639.5	.2779	70
80	4.289	647.7	3071	4.005	646.7	.2981	3.753	645.7	2896	80
90	4.388	653.7	3180	4.098	652.8	.3092	3.842	651.8	.3008	90
100	4.485	659.6	1.3286	4.190	658.7	1.3199	3.930	657.8	1.3117	100
110	4.581	665.4	.3389	4.281	664.6	.3303	4.016	663.8	.3221	110
120	4.676	671.1	.3489	4.371	670.4	.3404	4.101	669.6	.3323	120
130	4.770	676.8	3586	4.460	676.1	.3502	4.186	675.4	.3422	130
140	4.863	682.5	.3682	4.548	681.8	3598	4.269	681.2	.3519	140
150	4.956	688.1	1.3775	4.635	687.5	1.3692	4.352	686.9	1.3614	150
160	5.048	693.7	.3866	4.722	693.2	.3784	4.434	692.6	.3706	160
170	5.139	699.3	3956	4.808	698.8	.3874	4.515	698.2	.3797	170
180	5.230	704.9	.4044	4.893	704.4	.3963	4.596	703.9	.3886	180
190	5.320	710.5	.4131	4.978	710.0	.4050	4.677	709.5	.3974	190
200	5.410	716.1	1.4217	5.063	715.6	1.4136	4.757	715.2	1.4060	200
210	5.500	721.7	.4301	5.147	721.3	.4220	4.836	720.8	.4145	210
220	5.589	727.3	.4384	5.231	726.9	.4304	4.916	726.4	.4228	220
230	5.678	732.9	.4466	5.315	732.5	.4386	4.995	732.1	.4311	230
240	5.767	738.5	.4546	5.398	738.1	.4467	5.074	737.7	.4392	240
250	5.855	744.1	1.4625	5.482	743.8	1.4547	5.152	743.4	1.4472	250
260	5.943	749.8	.4705	5.565	749.4	.4626	5.230	749.0	.4551	260
280	6.119	761.1	.4860	5.730	760.7	.4781	5.386	760.4	.4707	280
300	6.294	772.4	.5011	5.894	772.1	.4933	5.541	771.8	.4859	300

Absolute pressure in lbs./in.² (Saturation temperature in italics.)

Temp. °F.	100 (56.05°) V	100 H	100 S	105 (58.67°) V	105 H	105 S	110 (61.21°) V	110 H	110 S	Temp. °F.
Sat.	*2.952*	*626.5*	*1.2356*	*2.817*	*627.0*	*1.2314*	*2.693*	*627.5*	*1.2275*	Sat.
70	3.068	636.0	1.2539	2.907	634.9	1.2464	2.761	633.7	1.2392	70
80	3.149	642.6	2661	2.985	641.5	2589	2.837	640.5	.2519	80
90	3.227	649.0	.2778	3.061	648.0	2708	2.910	647.0	.2640	90
100	3.304	655.2	1.2891	3.135	654.3	1.2822	2.981	653.4	1.2755	100
110	3.380	661.3	.2999	3.208	660.5	.2931	3.051	659.7	.2866	110
120	3.454	667.3	.3104	3.279	666.6	3037	3.120	665.8	.2972	120
130	3.527	673.3	.3206	3.350	672.6	.3139	3.188	671.9	3076	130
140	3.600	679.2	.3305	3.419	678.5	3239	3.255	677.8	.3176	140
150	3.672	685.0	1.3401	3.488	684.4	1.3336	3.321	683.7	1.3274	150
160	3.743	690.8	.3495	3.556	690.2	.3431	3.386	689.6	.3370	160
170	3.813	696.6	.3588	3.623	696.0	.3524	3.451	695.4	.3463	170
180	3.883	702.3	.3678	3.690	701.8	.3615	3.515	701.2	.3555	180
190	3.952	708.0	.3767	3.757	707.5	.3704	3.579	707.0	.3644	190
200	4.021	713.7	1.3854	3.823	713.3	1.3792	3.642	712.8	1.3732	200
210	4.090	719.4	.3940	3.888	719.0	.3878	3.705	718.5	.3819	210
220	4.158	725.1	.4024	3.954	724.7	.3963	3.768	724.3	.3904	220
230	4.226	730.8	.4108	4.019	730.4	.4046	3.830	730.0	.3988	230
240	4.294	736.5	.4190	4.083	736.1	.4129	3.892	735.7	.4070	240
250	4.361	742.2	1.4271	4.148	741.9	1.4210	3.954	741.5	1.4151	250
260	4.428	747.9	.4350	4.212	747.6	.4290	4.015	747.2	.4232	260
270	4.495	753.6	.4429	4.276	753.3	.4369	4.076	752.9	.4311	270
280	4.562	759.4	.4507	4.340	759.0	.4447	4.137	758.7	.4389	280
290	4.629	765.1	.4584	4.403	764.8	.4524	4.198	764.5	.4466	290
300	4.695	770.8	1.4660	4.466	770.5	1.4600	4.259	770.2	1.4543	300

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft. ^{3}/lb; H = heat content in Btu. /lb; S = entropy in Btu. /lb °F.)

Temp. °F.	Absolute pressure in lbs./in.^{2} (Saturation temperature in italics.) 90 *50.47°*			95 *53.35°*			100 *56.05°*			Temp. °F.
	V	H	S	V	H	S	V	H	S	
Sat.	*3.266*	*625.3*	*1.2446*	*3.101*	*625.9*	*1.2399*	*2.952*	*626.5*	*1.2356*	Sat.
50	......	......	......	......	......	......	......	......	......	50
60	3.353	631.8	1.2571	3.160	630.5	1.2489	2.985	629.3	1.2409	60
70	3.442	638.3	.2695	3.245	637.2	.2616	3.068	636.0	.2539	70
80	3.529	644.7	.2814	3.329	643.6	.2736	3.149	642.6	.2661	80
90	3.614	650.9	.2928	3.411	649.9	.2852	3.227	649.0	.2778	90
100	3.698	657.0	1.3038	3.491	656.1	1.2963	3.304	655.2	1.2891	100
110	3.780	663.0	.3144	3.570	662.1	.3070	3.380	661.3	.2999	110
120	3.862	668.9	.3247	3.647	668.1	.3174	3.454	667.3	.3104	120
130	3.942	674.7	.3347	3.724	674.0	.3275	3.527	673.3	.3206	130
140	4.021	680.5	.3444	3.799	679.8	.3373	3.600	679.2	.3305	140
150	4.100	686.3	1.3539	3.874	685.6	1.3469	3.672	685.0	1.3401	150
160	4.178	692.0	.3633	3.949	691.4	.3562	3.743	690.8	.3495	160
170	4.255	697.7	.3724	4.022	697.1	.3654	3.813	696.6	.3588	170
180	4.332	703.4	.3813	4.096	702.8	.3744	3.883	702.3	.3678	180
190	4.408	709.0	.3901	4.168	708.5	.3833	3.952	708.0	.3767	190
200	4.484	714.7	1.3988	4.241	714.2	1.3919	4.021	713.7	1.3854	200
210	4.560	720.4	.4073	4.313	719.9	.4005	4.090	719.4	.3940	210
220	4.635	726.0	.4157	4.384	725.6	.4089	4.158	725.1	.4024	220
230	4.710	731.7	.4239	4.455	731.3	.4172	4.226	730.8	.4108	230
240	4.785	737.3	.4321	4.526	736.9	.4254	4.294	736.5	.4190	240
250	4.859	743.0	1.4401	4.597	742.6	1.4334	4.361	742.2	1.4271	250
260	4.933	748.7	.4481	4.668	748.3	.4414	4.428	747.9	.4350	260
280	5.081	760.0	.4637	4.808	759.7	.4570	4.562	759.4	.4507	280
300	5.228	771.5	.4789	4.947	771.2	.4723	4.695	770.8	.4660	300
	115 *63.66°*			120 *66.02°*			125 *68.31°*			
Sat.	*2.580*	*628.0*	*1.2237*	*2.476*	*628.4*	*1.2201*	*2.380*	*628.8*	*1.2166*	Sat.
70	2.628	632.5	1.2323	2.505	631.3	1.2255	2.392	630.0	1.2189	70
80	2.701	639.4	.2451	2.576	638.3	.2386	2.461	637.2	.2322	80
90	2.772	646.0	.2574	2.645	645.0	.2510	2.528	644.0	.2448	90
100	2.841	652.5	1.2690	2.712	651.6	1.2628	2.593	650.7	1.2568	100
110	2.909	658.8	.2802	2.778	658.0	.2741	2.657	657.1	.2682	110
120	2.975	665.0	.2910	2.842	664.2	.2850	2.719	663.5	.2792	120
130	3.040	671.1	.3015	2.905	670.4	.2956	2.780	669.7	.2899	130
140	3.105	677.2	.3116	2.967	676.5	.3058	2.840	675.8	.3002	140
150	3.168	683.1	1.3215	3.029	682.5	1.3157	2.900	681.8	1.3102	150
160	3.231	689.0	.3311	3.089	688.4	.3254	2.958	687.8	.3199	160
170	3.294	694.9	.3405	3.149	694.3	.3348	3.016	693.7	.3294	170
180	3.355	700.7	.3497	3.209	700.2	.3441	3.074	699.6	.3387	180
190	3.417	706.5	.3587	3.268	706.0	.3531	3.131	705.5	.3478	190
200	3.477	712.3	1.3675	3.326	711.8	1.3620	3.187	711.3	1.3567	200
210	3.538	718.1	.3762	3.385	717.6	.3707	3.243	717.2	.3654	210
220	3.598	723.8	.3847	3.442	723.4	.3793	3.299	723.0	.3740	220
230	3.658	729.6	.3931	3.500	729.2	.3877	3.354	728.8	.3825	230
240	3.717	735.3	.4014	3.557	734.9	.3960	3.409	734.5	.3908	240
250	3.776	741.1	1.4096	3.614	740.7	1.4042	3.464	740.3	1.3990	250
260	3.835	746.8	.4176	3.671	746.5	.4123	3.519	746.1	.4071	260
270	3.894	752.6	.4256	3.727	752.2	.4202	3.573	751.9	.4151	270
280	3.952	758.4	.4334	3.783	758.0	.4281	3.627	757.7	.4230	280
290	4.011	764.1	.4411	3.839	763.8	.4359	3.681	763.5	.4308	290
300	4.069	769.9	1.4488	3.895	769.6	1.4435	3.735	769.3	1.4385	300

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft.^{3}/lb; H = heat content in Btu. /lb; S = entropy in Btu. /lb °F.)

Temp. °F.	125 (68.31°) V	125 H	125 S	130 (70.65°) V	130 H	130 S	135 (72.87°) V	135 H	135 S	Temp. °F.
Sat.	*2.380*	*628.8*	*1.2166*	*2.291*	*629.2*	*1.2132*	*2.209*	*629.6*	*1.2100*	Sat.
80	2.461	637.2	1.2322	2.355	636.0	1.2260	2.257	634.9	1.2199	80
90	2.528	644.0	.2448	2.421	643.0	.2388	2.321	642.0	.2329	90
100	2.593	650.7	1.2568	2.484	649.7	1.2509	2.382	648.8	1.2452	100
110	2.657	657.1	.2682	2.546	656.3	.2625	2.442	655.4	.2569	110
120	2.719	663.5	.2792	2.606	662.7	.2736	2.501	661.9	.2681	120
130	2.780	669.7	.2899	2.665	668.9	.2843	2.559	668.2	.2790	130
140	2.840	675.8	.3002	2.724	675.1	.2947	2.615	674.4	.2894	140
150	2.900	681.8	1.3102	2.781	681.2	1.3048	2.671	680.5	1.2996	150
160	2.958	687.8	.3199	2.838	687.2	.3146	2.726	686.6	.3094	160
170	3.016	693.7	.3294	2.894	693.2	.3241	2.780	692.6	.3191	170
180	3.074	699.6	.3387	2.949	699.1	.3335	2.834	698.6	.3284	180
190	3.131	705.5	3478	3.004	705.0	.3426	2.887	704.5	3376	190
200	3.187	711.3	1.3567	3.059	710.9	1.3516	2.940	710.4	1.3466	200
210	3.243	717.2	.3654	3.113	716.7	.3604	2.992	716.2	.3554	210
220	3.299	723.0	.3740	3.167	722.5	.3690	3.044	722.1	.3641	220
230	3.354	728.8	.3825	3.220	728.3	.3775	3.096	727.9	.3726	230
240	3.409	734.5	3908	3.273	734.1	3858	3.147	733.7	3810	240
250	3.464	740.3	1.3990	3.326	739.9	1.3941	3.198	739.6	1.3893	250
260	3.519	746.1	.4071	3.379	745.7	.4022	3.249	745.4	.3974	260
270	3.573	751.9	.4151	3.431	751.5	.4102	3.300	751.2	.4054	270
280	3.627	757.7	.4230	3.483	757.3	.4181	3.350	757.0	.4133	280
290	3.681	763.5	.4308	3.535	763.1	.4259	3.400	762.8	.4212	290
300	3.735	769.3	1.4385	3.587	769.0	1.4336	3.450	768.6	1.4289	300
320	3.842	780.9	.4536	3.690	780.6	.4487	3.550	780.3	.4441	320

Temp. °F.	150 (78.81°) V	150 H	150 S	160 (82.64°) V	160 H	160 S	170 (86.29°) V	170 H	170 S	Temp. °F.
Sat.	*1.994*	*630.5*	*1.2009*	*1.872*	*631.1*	*1.1952*	*1.764*	*631.8*	*1.1900*	Sat.
90	2.061	638.8	1.2161	1.914	636.6	1.2055	1.784	634.4	1.1952	90
100	2.118	645.9	1.2289	1.969	643.9	1.2186	1.837	641.9	1.2087	100
110	2.174	652.8	.2410	2.023	651.0	.2311	1.889	649.1	.2215	110
120	2.228	659.4	.2526	2.075	657.8	.2429	1.939	656.1	.2336	120
130	2.281	665.9	.2638	2.125	664.4	.2542	1.988	662.8	.2452	130
140	2.334	672.3	.2745	2.175	670.9	.2652	2.035	669.4	.2563	140
150	2.385	678.6	1.2849	2.224	677.2	1.2757	2.081	675.9	1.2669	150
160	2.435	684.8	.2949	2.272	683.5	.2859	2.127	682.3	.2773	160
170	2.485	690.9	.3047	2.319	689.7	.2958	2.172	688.5	.2873	170
180	2.534	696.9	.3142	2.365	695.8	.3054	2.216	694.7	.2971	180
190	2.583	702.9	.3236	2.411	701.9	.3148	2.260	700.8	.3066	190
200	2.631	708.9	1.3327	2.457	707.9	1.3240	2.303	706.9	1.3159	200
210	2.679	714.8	.3416	2.502	713.9	.3331	2.346	713.0	.3249	210
220	2.726	720.7	.3504	2.547	719.9	.3419	2.389	719.0	.3338	220
230	2.773	726.6	.3590	2.591	725.8	.3506	2.431	724.9	.3426	230
240	2.820	732.5	.3675	2.635	731.7	.3591	2.473	730.9	.3512	240
250	2.866	738.4	1.3758	2.679	737.6	1.3675	2.514	736.8	1.3596	250
260	2.912	744.3	.3840	2.723	743.5	.3757	2.555	742.8	.3679	260
270	2.958	750.1	.3921	2.766	749.4	.3838	2.596	748.7	.3761	270
280	3.004	756.0	.4001	2.809	755.3	.3919	2.637	754.6	.3841	280
290	3.049	761.8	.4079	2.852	761.2	.3998	2.678	760.5	.3921	290
300	3.095	767.7	1.4157	2.895	767.1	1.4076	2.718	766.4	1.3999	300
320	3.185	779.4	.4310	2.980	778.9	.4229	2.798	778.3	.4153	320
340	3.274	791.2	.4459	3.064	790.7	.4379	2.878	790.1	.4303	340

Absolute pressure in lbs./in.^{2} (Saturation temperature in italics.)

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft.^{3}/lb; H = heat content in Btu. /lb; S = entropy in Btu. /lb °F.)

Temp. °F.	Absolute pressure in lbs./in.² (Saturation temperature in italics.)									Temp. °F.
	140 *74.79°*			145 *76.83°*			150 *78.81°*			
	V	H	S	V	H	S	V	H	S	
Sat.	*2.132*	*629.9*	*1.2068*	*2.061*	*630.2*	*1.2033*	*1.994*	*630.5*	*1.2000*	Sat.
80	2.166	633.8	1.2140	2.080	632.6	1.2082	2.001	631.4	1.2025	80
90	2.228	640.9	.2272	2.141	639.9	.2216	2.061	638.8	.2161	90
100	2.288	647.8	1.2396	2.200	646.9	1.2342	2.118	645.9	1.2289	100
110	2.347	654.5	.2515	2.257	653.6	.2462	2.174	652.8	.2410	110
120	2.404	661.1	.2628	2.313	660.2	.2577	2.228	659.4	.2526	120
130	2.460	667.4	.2738	2.368	666.7	.2687	2.281	665.9	.2638	130
140	2.515	673.7	.2843	2.421	673.0	.2793	2.334	672.3	.2745	140
150	2.569	679.9	1.2945	2.474	679.2	1.2896	2.385	678.6	1.2849	150
160	2.622	686.0	.3045	2.526	685.4	.2996	2.435	684.8	.2949	160
170	2.675	692.0	.3141	2.577	691.4	.3093	2.485	690.9	.3047	170
180	2.727	698.0	.3236	2.627	697.5	.3188	2.534	696.9	.3142	180
190	2.779	704.0	.3328	2.677	703.4	.3281	2.583	702.9	.3236	190
200	2.830	709.9	1.3418	2.727	709.4	1.3372	2.631	708.9	1.3327	200
210	2.880	715.8	.3507	2.776	715.3	.3461	2.679	714.8	.3416	210
220	2.931	721.6	.3594	2.825	721.2	.3548	2.726	720.7	.3504	220
230	2.981	727.5	.3679	2.873	727.1	.3634	2.773	726.6	.3590	230
240	3.030	733.3	.3763	2.921	732.9	.3718	2.820	732.5	.3675	240
250	3.080	739.2	1.3846	2.969	738.8	1.3801	2.866	738.4	1.3758	250
260	3.129	745.0	.3928	3.017	744.6	.3883	2.912	744.3	.3840	260
270	3.179	750.8	.4008	3.064	750.5	.3964	2.958	750.1	.3921	270
280	3.227	756.7	.4088	3.111	756.3	.4043	3.004	756.0	.4001	280
290	3.276	762.5	.4166	3.158	762.2	.4122	3.049	761.8	.4079	290
300	3.323	768.3	1.4243	3.205	768.0	1.4199	3.095	767.7	1.4157	300
320	3.420	780.0	.4395	3.298	779.7	.4352	3.185	779.4	.4310	320
	180 *89.78°*			190 *93.15°*			200 *96.34°*			
Sat.	*1.667*	*632.0*	*1.1850*	*1.581*	*632.4*	*1.1806*	*1.502*	*632.7*	*1.1756*	Sat.
90	1.668	632.2	1.1853	....	....	....	....	....	....	90
100	1.720	639.9	1.1992	1.615	637.8	1.1899	1.520	635.6	1.1809	100
110	1.770	647.3	.2123	1.663	645.4	.2034	1.567	643.4	.1947	110
120	1.818	654.4	.2247	1.710	652.6	.2160	1.612	650.9	.2077	120
130	1.865	661.3	.2364	1.755	659.7	.2281	1.656	658.1	.2200	130
140	1.910	668.0	.2477	1.799	666.5	.2396	1.698	665.0	.2317	140
150	1.955	674.6	1.2586	1.842	673.2	1.2506	1.740	671.8	1.2429	150
160	1.999	681.0	.2691	1.884	679.7	.2612	1.780	678.4	.2537	160
170	2.042	687.3	.2792	1.925	686.1	.2715	1.820	684.9	.2641	170
180	2.084	693.6	.2891	1.966	692.5	.2815	1.859	691.3	.2742	180
190	2.126	699.8	.2987	2.005	698.7	.2912	1.897	697.7	.2840	190
200	2.167	705.9	1.3081	2.045	704.9	1.3007	1.935	703.9	1.2935	200
210	2.208	712.0	.3172	2.084	711.1	.3099	1.972	710.1	.3029	210
220	2.248	718.1	.3262	2.123	717.2	.3189	2.009	716.3	.3120	220
230	2.288	724.1	.3350	2.161	723.2	.3278	2.046	722.4	.3209	230
240	2.328	730.1	.3436	2.199	729.3	.3365	2.082	728.4	.3296	240
250	2.367	736.1	1.3521	2.236	735.3	1.3450	2.118	734.5	1.3382	250
260	2.407	742.0	.3605	2.274	741.3	.3534	2.154	740.5	.3467	260
270	2.446	748.0	.3687	2.311	747.3	.3617	2.189	746.5	.3550	270
280	2.484	753.9	.3768	2.348	753.2	.3698	2.225	752.5	.3631	280
290	2.523	759.9	.3847	2.384	759.2	.3778	2.260	758.5	.3712	290
300	2.561	765.8	1.3926	2.421	765.2	1.3857	2.295	764.5	1.3791	300
320	2.637	777.7	.4081	2.493	777.1	.4012	2.364	776.5	.3947	320
340	2.713	789.6	.4231	2.565	789.0	.4163	2.432	788.5	.4099	340

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Continued.)

(V = volume in ft.³/lb; H = heat content in Btu. /lb; S = entropy in Btu. /lb °F.)

Temp. °C.	200 (96.34°) V	200 H	200 S	210 (99.43°) V	210 H	210 S	220 (102.42°) V	220 H	220 S	Temp. °F.
Sat.	*1.502*	*632.7*	*1.1756*	*1.431*	*633.0*	*1.1713*	*1.367*	*633.2*	*1.1671*	Sat.
110	1.567	643.4	1.1947	1.480	641.5	1.1863	1.400	639.4	1.1781	110
120	1.612	650.9	.2077	1.524	649.1	.1996	1.443	647.3	.1917	120
130	1.656	658.1	.2200	1.566	656.4	.2121	1.485	654.8	.2045	130
140	1.698	665.0	.2317	1.608	663.5	.2240	1.525	662.0	.2167	140
150	1.740	671.8	1.2429	1.648	670.4	1.2354	1.564	669.0	1.2281	150
160	1.780	678.4	.2537	1.687	677.1	.2464	1.601	675.8	.2394	160
170	1.820	684.9	.2641	1.725	683.7	.2569	1.638	682.5	.2501	170
180	1.859	691.3	.2742	1.762	690.2	.2672	1.675	689.1	.2604	180
190	1.897	697.7	.2840	1.799	696.6	.2771	1.710	695.5	.2704	190
200	1.935	703.9	1.2935	1.836	702.9	1.2867	1.745	701.9	1.2801	200
210	1.972	710.1	.3029	1.872	709.2	.2961	1.780	708.2	.2896	210
220	2.009	716.3	.3120	1.907	715.3	.3053	1.814	714.4	.2989	220
230	2.046	722.4	.3209	1.942	721.5	.3143	1.848	720.6	.3079	230
240	2.082	728.4	.3296	1.977	727.6	.3231	1.881	726.8	.3168	240
250	2.118	734.5	1.3382	2.011	733.7	1.3317	1.914	732.9	1.3255	250
260	2.154	740.5	.3467	2.046	739.8	.3402	1.947	739.0	.3340	260
270	2.189	746.5	.3550	2.080	745.8	.3486	1.980	745.1	.3424	270
280	2.225	752.5	.3631	2.113	751.8	.3568	2.012	751.1	.3507	280
290	2.260	758.5	.3712	2.147	757.9	.3649	2.044	757.2	.3588	290
300	2.295	764.5	1.3791	2.180	763.9	1.3728	2.076	763.2	1.3668	300
320	2.364	776.5	.3947	2.246	775.9	.3884	2.140	775.3	.3825	320
340	2.432	788.5	.4099	2.312	787.9	.4037	2.203	787.4	.3978	340
360	2.500	800.5	.4247	2.377	800.0	.4186	2.265	799.5	.4127	360
380	2.568	812.5	.4392	2.442	812.0	.4331	2.327	811.6	.4273	380

Temp. °C.	250 (110.80°) V	250 H	250 S	260 (113.42°) V	260 H	260 S	270 (115.97°) V	270 H	270 S	Temp. °F.
Sat.	*1.202*	*633.8*	*1.1566*	*1.155*	*633.9*	*1.1518*	*1.112*	*633.9*	*1.1483*	Sat.
120	1.240	641.5	1.1690	1.182	639.5	1.1617	1.128	637.5	1.1544	120
130	1.278	649.6	.1827	1.220	647.8	.1757	1.166	645.9	.1689	130
140	1.316	657.2	.1956	1.257	655.6	.1889	1.202	653.9	.1823	140
150	1.352	664.6	1.2078	1.292	663.1	1.2014	1.236	661.6	1.1950	150
160	1.386	671.8	.2195	1.326	670.4	.2132	1.269	669.0	.2071	160
170	1.420	678.7	.2306	1.359	677.5	.2245	1.302	676.2	.2185	170
180	1.453	685.5	.2414	1.391	684.4	.2354	1.333	683.2	.2296	180
190	1.486	692.2	.2517	1.422	691.1	.2458	1.364	690.0	.2401	190
200	1.518	698.8	1.2617	1.453	697.7	1.2560	1.394	696.7	1.2504	200
210	1.549	705.3	.2715	1.484	704.3	.2658	1.423	703.3	.2603	210
220	1.580	711.7	.2810	1.514	710.7	.2754	1.452	709.8	.2700	220
230	1.610	718.0	.2902	1.543	717.1	.2847	1.481	716.2	.2794	230
240	1.640	724.3	.2993	1.572	723.4	.2938	1.509	722.6	.2885	240
250	1.670	730.5	1.3081	1.601	729.7	1.3027	1.537	728.9	1.2975	250
260	1.699	736.7	.3168	1.630	736.0	.3115	1.565	735.2	.3063	260
270	1.729	742.9	.3253	1.658	742.2	.3200	1.592	741.4	.3149	270
280	1.758	749.1	.3337	1.686	748.4	.3285	1.620	747.7	.3234	280
290	1.786	755.2	.3420	1.714	754.5	.3367	1.646	753.9	.3317	290
300	1.815	761.3	1.3501	1.741	760.7	1.3449	1.673	760.0	1.3399	300
320	1.872	773.5	.3659	1.796	772.9	.3608	1.726	772.3	.3559	320
340	1.928	785.7	.3814	1.850	785.2	.3763	1.778	784.6	.3714	340
360	1.983	797.9	.3964	1.904	797.4	.3914	1.830	796.9	.3866	360
380	2.038	810.1	.4111	1.957	809.6	.4062	1.881	809.1	.4014	380
400	2.093	822.3	1.4255	2.009	821.9	1.4206	1.932	821.4	1.4158	400

Absolute pressure in lbs./in.² (Saturation temperature in italics.)

TABLE 10.—BUREAU OF STANDARDS TABLES OF PROPERTIES OF SUPERHEATED AMMONIA VAPOR.—(Concluded.)

(V = volume in ft.³/lb; H = heat content in B.t.u. /lb; S = entropy in B.t.u. /lb °F.)

Temp. °F.	Absolute pressure in lbs./in.² (Saturation temperature in italics.)									Temp. °F.
	230 *105.30°*			240 *108.09°*			250 *110.80°*			
	V	H	S	V	H	S	V	H	S	
Sat.	*1.307*	*633.4*	*1.1651*	*1.253*	*633.6*	*1.1592*	*1.202*	*633.8*	*1.1655*	*Sat.*
110	1.328	637.4	1.1700	1.261	635.3	1.1621	....	....	....	110
120	1.370	645.4	.1840	1.302	643.5	.1764	1.240	641.5	1.1690	120
130	1.410	653.1	.1971	1.342	651.3	.1898	1.278	649.6	.1827	130
140	1.449	660.4	.2095	1.380	658.8	.2025	1.316	657.2	.1956	140
150	1.487	667.6	1.2213	1.416	666.1	1.2145	1.352	664.6	1.2078	150
160	1.524	674.5	.2325	1.452	673.1	.2259	1.386	671.8	.2195	160
170	1.559	681.3	.2434	1.487	680.0	.2369	1.420	678.7	.2306	170
180	1.594	687.9	.2538	1.521	686.7	.2475	1.453	685.5	.2414	180
190	1.629	694.4	.2640	1.554	693.3	.2577	1.486	692.2	.2517	190
200	1.663	700.9	1.2738	1.587	699.8	1.2677	1.518	698.8	1.2617	200
210	1.696	707.2	.2834	1.619	706.2	.2773	1.549	705.3	.2715	210
220	1.729	713.5	.2927	1.651	712.6	.2867	1.580	711.7	.2810	220
230	1.762	719.8	.3018	1.683	718.9	.2959	1.610	718.0	.2902	230
240	1.794	726.0	.3107	1.714	725.1	.3049	1.640	724.3	.2993	240
250	1.826	732.1	1.3195	1.745	731.3	1.3137	1.670	730.5	1.3081	250
260	1.857	738.3	.3281	1.775	737.5	.3224	1.699	736.7	.3168	260
270	1.889	744.4	.3365	1.805	743.6	.3308	1.729	742.9	.3253	270
280	1.920	750.5	.3448	1.835	749.8	.3392	1.758	749.1	.3337	280
290	1.951	756.5	.3530	1.865	755.9	.3474	1.786	755.2	.3420	290
300	1.982	762.6	1.3610	1.895	762.0	1.3554	1.815	761.3	1.3501	300
320	2.043	774.7	.3767	1.954	774.1	.3712	1.872	773.5	.3659	320
340	2.103	786.8	.3921	2.012	786.3	.3866	1.928	785.7	.3814	340
360	2.163	798.9	.4070	2.069	798.4	.4016	1.983	797.9	.3964	360
380	2.222	811.1	.4217	2.126	810.6	.4163	2.038	810.1	.4111	380
	280 *118.45°*			290 *120.86°*			300 *123.21°*			
Sat.	*1.072*	*634.0*	*1.1449*	*1.034*	*634.0*	*1.1415*	*0.999*	*634.0*	*1.1385*	*Sat.*
120	1.078	635.4	1.1473	....	....	....	....	....	....	120
130	1.115	644.0	.1621	1.068	642.1	1.1554	1.023	640.1	1.1487	130
140	1.151	652.2	.1759	1.103	650.5	.1695	1.058	648.7	.1632	140
150	1.184	660.1	1.1888	1.136	658.5	1.1827	1.091	656.9	1.1767	150
160	1.217	667.6	.2011	1.168	666.1	.1952	1.123	664.7	.1894	160
170	1.249	674.9	.2127	1.199	673.5	.2070	1.153	672.2	.2014	170
180	1.279	681.9	.2239	1.229	680.7	.2183	1.183	679.5	.2129	180
190	1.309	688.9	.2346	1.259	687.7	.2292	1.211	686.5	.2239	190
200	1.339	695.6	1.2449	1.287	694.6	1.2396	1.239	693.5	1.2344	200
210	1.367	702.3	.2550	1.315	701.3	.2497	1.267	700.3	.2447	210
220	1.396	708.8	.2647	1.343	707.9	.2596	1.294	706.9	.2546	220
230	1.424	715.3	.2742	1.370	714.4	.2691	1.320	713.5	.2642	230
240	1.451	721.8	.2834	1.397	720.9	.2784	1.346	720.0	.2736	240
250	1.478	728.1	1.2924	1.423	727.3	1.2875	1.372	726.5	1.2827	250
260	1.505	734.4	.3013	1.449	733.7	.2964	1.397	732.9	.2917	260
270	1.532	740.7	.3099	1.475	740.0	.3051	1.422	739.2	.3004	270
280	1.558	747.0	.3184	1.501	746.3	.3137	1.447	745.5	.3090	280
290	1.584	753.2	.3268	1.526	752.5	.3221	1.472	751.8	.3175	290
300	1.610	759.4	1.3350	1.551	758.7	1.3303	1.496	758.1	1.3257	300
320	1.661	771.7	.3511	1.601	771.1	.3464	1.544	770.5	.3419	320
340	1.712	784.0	.3667	1.650	783.5	.3621	1.592	782.9	.3576	340
360	1.762	796.3	.3819	1.698	795.8	.3773	1.639	795.3	.3729	360
380	1.811	808.7	.3967	1.747	808.2	.3922	1.686	807.7	.3878	380
400	1.861	821.0	1.4112	1.794	820.5	1.4067	1.732	820.1	1.4024	400

TABLE 11.—F12, DICHLORODIFLUOROMETHANE (CCl_2F_2).
PROPERTIES OF SATURATED VAPOR.

Temp.	Pressure		Volume		Density		Heat content from − 40°			Entropy from − 40°		Temp.
°F t	Abs. lb./in.² p	Gage lb./in.² p_d	Liquid ft.³/lb. v_f	Vapor ft.³/lb. v_g	Liquid lb./ft.³ $1/v_f$	Vapor lb./ft.³ $1/v_g$	Liquid Btu./lb. h_f	Latent Btu./lb. h	Vapor Btu./lb. h_g	Liquid Btu./lb.°F s_f	Vapor Btu./lb.°F s_g	°F t
−40	9.32	10.92*	0.0106	3.911	94.58	0.2557	0	73.50	73.50	0	0.17517	−40
−38	9.82	9.91*	.0106	3.727	94.39	.2683	0.40	73.34	73.74	0.00094	.17490	−38
−36	10.34	8.87*	.0106	3.553	94.20	.2815	0.81	73.17	73.98	.00188	.17463	−36
−34	10.87	7.80*	.0106	3.389	93.99	.2951	1.21	73.01	74.22	.00282	.17438	−34
−32	11.43	6.66*	.0107	3.234	93.79	.3092	1.62	72.84	74.46	.00376	.17412	−32
−30	12.02	5.45*	0.0107	3.088	93.59	0.3238	2.03	72.67	74.70	0.00471	0.17387	−30
−28	12.62	4.23*	.0107	2.950	93.39	.3390	2.44	72.50	74.94	.00565	.17364	−28
−26	13.26	2.93*	.0107	2.820	93.18	.3546	2.85	72.33	75.18	.00659	.17340	−26
−24	13.90	1.63*	.0108	2.698	92.98	.3706	3.25	72.16	75.41	.00753	.17317	−24
−22	14.58	0.24*	.0108	2.583	92.78	.3871	3.66	71.98	75.64	.00846	.17296	−22
−20	15.28	0.58	0.0108	2.474	92.58	0.4042	4.07	71.80	75.87	0.00940	0.17275	−20
−18	16.01	1.31	.0108	2.370	92.38	.4219	4.48	71.63	76.11	.01033	.17253	−18
−16	16.77	2.07	.0108	2.271	92.18	.4403	4.89	71.45	76.34	.01126	.17232	−16
−14	17.55	2.85	.0109	2.177	91.97	.4593	5.30	71.27	76.57	.01218	.17212	−14
−12	18.37	3.67	.0109	2.088	91.77	.4789	5.72	71.09	76.81	.01310	.17194	−12
−10	19.20	4.50	0.0109	2.003	91.57	0.4993	6.14	70.91	77.05	0.01403	0.17175	−10
−8	20.08	5.38	.0109	1.922	91.35	.5203	6.57	70.72	77.29	.01496	.17158	−8
−6	20.98	6.28	.0110	1.845	91.14	.5420	6.99	70.53	77.52	.01589	.17140	−6
−4	21.91	7.21	.0110	1.772	90.93	.5644	7.41	70.34	77.75	.01682	.17123	−4
−2	22.87	8.17	.0110	1.703	90.72	.5872	7.83	70.15	77.98	.01775	.17107	−2
0	23.87	9.17	0.0110	1.637	90.52	0.6109	8.25	69.96	78.21	0.01869	0.17091	0
2	24.89	10.19	.0110	1.574	90.31	.6352	8.67	69.77	78.44	.01961	.17075	2
4	25.96	11.26	.0111	1.514	90.11	.6606	9.10	69.57	78.67	.02052	.17060	4
5†	26.51	11.81	.0111	1.485	90.00	.6735	9.32	69.47	78.79	.02097	.17052	5†
6	27.05	12.35	.0111	1.457	89.88	.6864	9.53	69.37	78.90	.02143	.17045	6
8	28.18	13.48	.0111	1.403	89.68	.7129	9.96	69.17	79.13	.02235	.17030	8
10	29.35	14.65	0.0112	1.351	89.45	0.7402	10.39	68.97	79.36	0.02328	0.17015	10
12	30.56	15.86	.0112	1.301	89.24	.7687	10.82	68.77	79.59	.02419	.17001	12
14	31.80	17.10	.0112	1.253	89.03	.7981	11.26	68.56	79.82	.02510	.16987	14
16	33.08	18.38	.0112	1.207	88.81	.8288	11.70	68.35	80.05	.02601	.16974	16
18	34.40	19.70	.0113	1.163	88.58	.8598	12.12	68.15	80.27	.02692	.16961	18
20	35.75	21.05	0.0113	1.121	88.37	0.8921	12.55	67.94	80.49	0.02783	0.16949	20
22	37.15	22.45	.0113	1.081	88.13	.9251	13.00	67.72	80.72	.02873	.16938	22
24	38.58	23.88	.0113	1.043	87.91	.9588	13.44	67.51	80.95	.02963	.16926	24
26	40.07	25.37	.0114	1.007	87.68	.9930	13.88	67.29	81.17	.03053	.16913	26
28	41.59	26.89	.0114	0.973	87.47	1.028	14.32	67.07	81.39	.03143	.16900	28
30	43.16	28.46	0.0115	0.939	87.24	1.065	14.76	66.85	81.61	0.03233	0.16887	30
32	44.77	30.07	.0115	.908	87.02	1.102	15.21	66.62	81.83	.03323	.16876	32
34	46.42	31.72	.0115	.877	86.78	1.140	15.65	66.40	82.05	.03413	.16865	34
36	48.13	33.43	.0116	.848	86.55	1.180	16.10	66.17	82.27	.03502	.16854	36
38	49.88	35.18	.0116	.819	86.33	1.221	16.55	65.94	82.49	.03591	.16843	38
40	51.68	36.98	0.0116	0.792	86.10	1.263	17.00	65.71	82.71	0.03680	0.16833	40
42	53.51	38.81	.0116	.767	85.88	1.304	17.46	65.47	82.93	.03770	.16823	42
44	55.40	40.70	.0117	.742	85.66	1.349	17.91	65.24	83.15	.03859	.16813	44
46	57.35	42.65	.0117	.718	85.43	1.393	18.36	65.00	83.36	.03948	.16803	46
48	59.35	44.65	.0117	.695	85.19	1.438	18.82	64.74	83.57	.04037	.16794	48
50	61.39	46.69	0.0118	0.673	84.94	1.485	19.27	64.51	83.78	0.04126	0.16785	50
52	63.49	48.79	.0118	.652	84.71	1.534	19.72	64.27	83.99	.04215	.16776	52
54	65.63	50.93	.0118	.632	84.50	1.583	20.18	64.02	84.20	.04304	.16767	54
56	67.84	53.14	.0119	.612	84.28	1.633	20.64	63.77	84.41	.04392	.16758	56
58	70.10	55.40	.0119	.593	84.04	1.686	21.11	63.51	84.62	.04480	.16749	58
60	72.41	57.71	0.0119	0.575	83.78	1.740	21.57	63.25	84.82	0.04568	0.16741	60
62	74.77	60.07	.0120	.557	83.57	1.795	22.03	62.99	85.02	.04657	.16733	62
64	77.20	62.50	.0120	.540	83.34	1.851	22.49	62.73	85.22	.04745	.16725	64
66	79.67	64.97	.0120	.524	83.10	1.909	22.95	62.47	85.42	.04833	.16717	66
68	82.24	67.54	.0121	.508	82.86	1.968	23.42	62.20	85.62	.04921	.16709	68
70	84.82	70.12	0.0121	0.493	82.60	2.028	23.90	61.92	85.82	0.05009	0.16701	70
72	87.50	72.80	.0121	.479	82.37	2.090	24.37	61.65	86.02	.05097	.16693	72
74	90.20	75.50	.0122	.464	82.12	2.153	24.84	61.38	86.22	.05185	.16685	74
76	93.00	78.30	.0122	.451	81.87	2.218	25.32	61.10	86.42	.05272	.16677	76
78	95.85	81.15	.0123	.438	81.62	2.284	25.80	60.81	86.61	.05359	.16669	78
80	98.76	84.06	0.0123	0.425	81.39	2.353	26.28	60.52	86.80	0.05446	0.16662	80

* Inches of mercury below one atmosphere.
† Standard ton temperatures.

Table 11.—F12, Dichlorodifluoromethane (CCl_2F_2). Properties of Saturated Vapor.—(Concluded.)

Temp.	Pressure		Volume		Density		Heat content from − 40°			Entropy from − 40°		Temp.
°F t	Abs. lb./in.² p	Gage lb./in.² p_d	Liquid ft.³/lb. v_f	Vapor ft.³/lb. v_g	Liquid lb./ft.³ $1/v_f$	Vapor lb./ft.³ $1/v_g$	Liquid Btu./lb. h_f	Latent Btu./lb. h	Vapor Btu./lb. h_g	Liquid Btu./lb.°F s_f	Vapor Btu./lb.°F s_g	°F t
80	98.76	84.06	0.0123	0.425	81.39	2.353	26.28	60.52	86.80	0.05446	0.16662	80
82	101.7	87.00	.0123	.413	81.12	2.423	26.76	60.23	86.99	.05534	.16655	82
84	104.8	90.1	.0124	.401	80.87	2.495	27.24	59.94	87.18	.05621	.16648	84
86†	**107.9**	**93.2**	**.0124**	**.389**	**80.63**	**2.569**	**27.72**	**69.65**	**87.37**	**.05708**	**.16640**	**86†**
88	111.1	96.4	.0124	.378	80.37	2.645	28.21	59.35	87.56	.05795	.16632	88
90	114.3	99.6	0.0125	0.368	80.11	2.721	28.70	59.04	87.74	0.05882	0.16624	90
92	117.7	103.0	.0125	.357	79.86	2.799	29.19	58.73	87.92	.05969	.16616	92
94	121.0	106.3	.0126	.347	79.60	2.880	29.68	58.42	88.10	.06056	.16608	94
96	124.5	109.8	.0126	.338	79.32	2.963	30.18	58.10	88.28	.06143	.16600	96
98	128.0	113.3	.0126	.328	79.06	3.048	30.67	57.78	88.45	.06230	.16592	98
100	131.6	116.9	0.0127	0.319	78.80	3.135	31.16	57.46	88.62	0.06316	0.16584	100
102	135.3	120.6	.0127	.310	78.54	3.224	31.65	57.14	88.79	.06403	.16576	102
104	139.0	124.3	.0128	.302	78.27	3.316	32.15	56.80	88.95	.06490	.16568	104
106	142.8	128.1	.0128	.293	78.00	3.411	32.65	56.46	89.11	.06577	.16560	106
108	146.8	132.1	.0129	.285	77.73	3.509	33.15	56.12	89.27	.06663	.16551	108
110	150.7	136.0	0.0129	0.277	77.46	3.610	33.65	55.78	89.43	0.06749	0.16542	110
112	154.8	140.1	.0130	.269	77.18	3.714	34.15	55.43	89.58	.06836	.16533	112
114	158.9	144.2	.0130	.262	76.89	3.823	34.65	55.08	89.73	.06922	.16524	114
116	163.1	148.4	.0131	.254	76.60	3.934	35.15	54.72	89.87	.07008	.16515	116
118	167.4	152.7	0131	.247	76.32	4.049	35.65	54.36	90.01	.07094	.16505	118
120	171.8	157.1	0.0132	0.240	76.02	4.167	36.16	53.99	90.15	0.07180	0.16495	120
122	176.2	161.5	.0132	.233	75.72	4.288	36.66	53.62	90.28	.07266	.16484	122
124	180.8	166.1	.0133	.277	75.40	4.413	37.16	53.24	90.40	.07352	.16473	124
126	185.4	170.7	.0133	.220	75.10	4.541	37.67	52.85	90.52	.07437	.16462	126
128	190.1	175.4	.0134	.214	74.78	4.673	38.18	52.46	90.64	.07522	.16450	128
130	194.9	180.2	0.0134	0.208	74.46	4.808	38.69	52.07	90.76	0.07607	0.16438	130
132	199.8	185.1	.0135	.202	74.13	4.948	39.19	51.67	90.86	.07691	.16425	132
134	204.8	190.1	.0135	.196	73.81	5.094	39.70	51.26	90.96	.07775	.16411	134
136	209.9	195.2	.0136	.191	73.46	5.247	40.21	50.85	91.06	.07858	.16396	136
138	215.0	200.3	.0137	.185	73.10	5.405	40.72	50.43	91.15	.07941	.16380	138
140	220.2	205.5	0.0138	0.180	72.73	5.571	41.24	50.00	91.24	0.08024	0.16363	140

Table 12.—F12, Dichlorodifluoromethane (CCl_2F_2).
Properties of Superheated Vapor.

Temp. °F.	Abs. Pressure 8 lb./in.^{2} Gage Pressure 13.6 in. vac. (Sat'n. Temp. −45.8° F.)			Abs. Pressure 9 lb./in.^{2} Gage Pressure 11.6 in. vac. (Sat'n. Temp. −41.4° F.)			Abs. Pressure 10 lb./in.^{2} Gage Pressure 9.6 in. vac. (Sat'n. Temp. −37.3° F.)			Abs. Pressure 11 lb./in.^{2} Gage Pressure 7.5 in. vac. (Sat'n. Temp. −33.5° F.)		
t	V	H	S	V	H	S	V	H	S	V	H	S
(at sat'n)	(4.502)	(72.80)	(0.17596)	(4.036)	(73.32)	(0.17555)	(3.658)	(73.80)	(0.17480)	(3.356)	(74.27)	(0.17438)
−40	4.569	73.56	0.17777	4.050	73.51	0.17576	......	......	......	......	......	......
−30	4.684	74.87	.18085	4.152	74.83	.17884	3.728	74.77	0.17704	3.383	74.73	0.17540
−20	4.799	76.20	.18390	4.255	76.15	.18188	3.821	76.11	.18008	3.467	76.06	.17845
−10	4.914	77.54	.18691	4.357	77.49	.18490	3.913	77.46	.18310	3.551	77.40	.18146
0	5.028	78.89	0.18991	4.460	78.84	0.18791	4.006	78.81	0.18611	3.635	78.75	0.18448
10	5.142	80.26	.19284	4.562	80.20	.19084	4.098	80.18	.18905	3.719	80.12	.18742
20	5.257	81.64	.19574	4.663	81.58	.19374	4.189	81.56	.19194	3.802	81.50	.19032
30	5.370	83.02	.19860	4.766	82.98	.19661	4.280	82.94	.19482	3.887	82.90	.19320
40	5.484	84.43	.20143	4.867	84.39	.19945	4.371	84.35	.19766	3.971	84.31	.19605
50	5.598	85.85	0.20425	4.969	85.80	0.20227	4.463	85.77	0.20047	4.055	85.73	0.19887
60	5.711	87.27	.20703	5.071	87.24	.20505	4.556	87.19	.20326	4.138	87.16	.20165
70	5.824	88.72	.20977	5.171	88.68	.20779	4.648	88.64	.20601	4.221	88.61	.20440
80	5.938	90.18	.21250	5.272	90.13	.21051	4.740	90.11	.20874	4.304	90.07	.20713
90	6.051	91.64	.21519	5.374	91.60	.21321	4.832	91.58	.21144	4.388	91.54	.20984
100	6.165	93.13	0.21786	5.475	93.09	0.21588	4.923	93.05	0.21411	4.471	93.03	0.21251
110	6.278	94.63	.22051	5.576	94.59	.21853	5.015	94.56	.21676	4.553	94.52	.21515
120	6.391	96.13	.22314	5.677	96.10	.22116	5.107	96.07	.21940	4.636	96.03	.21778
130	6.504	97.64	.22573	5.778	97.63	.22375	5.198	97.59	.22199	4.718	97.56	.22037
140	6.617	99.18	.22831	5.879	99.16	.22634	5.289	99.14	.22458	4.800	99.09	.22296
150	6.730	100.73	0.23087	5.979	100.70	0.22889	5.379	100.66	0.22713	4.882	100.63	0.22551
160	6.843	102.29	.23340	6.080	102.26	.23143	5.470	102.24	.22967	4.965	102.20	.22805
170	6.955	103.87	.23591	6.180	103.84	.23394	5.560	103.81	.23218	5.047	103.78	.23057
180	7.068	105.44	.23842	6.280	105.43	.23645	5.650	105.40	.23469	5.130	105.37	.23308
190	7.181	107.05	.24090	6.380	107.03	.23893	5.740	107.00	.23717	5.214	106.97	.23557
200	7.294	108.67	0.24337	6.481	108.64	0.24140	5.831	108.63	0.23963	5.297	108.58	0.23804
210	7.407	110.28	.24581	6.581	110.26	.24384	5.921	110.25	.24208	5.379	110.21	.24049
220	7.520	111.93	.24825	6.682	111.90	.24628	6.011	111.88	.24451	5.462	111.85	.24291
230	7.633	113.57	.25066	6.782	113.55	.24868	6.101	113.53	.24692	5.544	113.50	.24532
240	......	......	......	......	......	......	......	......	......	5.626	115.18	.24773

Temp. °F.	Abs. Pressure 12 lb./in.^{2} Gage Pressure 5.5 in. vac. (Sat'n. Temp. −30.0° F.)			Abs. Pressure 13 lb./in.^{2} Gage Pressure 3.5 in. vac. (Sat'n. Temp. −26.8° F.)			Abs. Pressure 14 lb./in.^{2} Gage Pressure 1.4 in. vac. (Sat'n. Temp. −23.7° F.)			Abs. Pressure 15 lb./in.^{2} Gage Pressure 0.3 lb./in.^{2} (Sat'n. Temp. −20.8° F.)		
(at sat'n)	(3.093)	(74.69)	(0.17389)	(2.876)	(75.08)	(0.17350)	(2.677)	(75.46)	(0.17314)	(2.518)	(75.75)	(0.17282)
−30	3.093	74.69	0.17389	......	......	......	......	......	......	......	......	......
−20	3.172	76.02	.17695	2.920	75.98	0.17556	2.706	75.94	0.17427	2.521	75.89	0.17307
−10	3.250	77.37	.17998	2.992	77.32	.17859	2.773	77.28	.17731	2.583	77.23	.17611
0	3.328	78.73	0.18299	3.064	78.69	0.18160	2.841	78.64	0.18032	2.646	78.59	0.17913
10	3.405	80.10	.18594	3.136	80.05	.18455	2.908	80.01	.18328	2.708	79.97	.18208
20	3.483	81.48	.18884	3.207	81.43	.18746	2.974	81.40	.18618	2.771	81.37	.18499
30	3.560	82.87	.19173	3.278	82.83	.19034	3.041	82.80	.18907	2.833	82.77	.18788
40	3.637	84.28	.19458	3.349	84.23	.19319	3.107	84.21	.19192	2.895	84.18	.19074
50	3.714	85.71	0.19739	3.420	85.66	0.19601	3.173	85.63	0.19475	2.957	85.60	0.19357
60	3.790	87.14	.20018	3.491	87.10	.19880	3.239	87.06	.19753	3.019	87.03	.19635
70	3.867	88.59	.20293	3.562	88.54	.20156	3.303	88.51	.20028	3.081	88.48	.19911
80	3.943	90.05	.20566	3.632	90.01	.20428	3.369	89.97	.20302	3.143	89.94	.20185
90	4.019	91.52	.20836	3.703	91.48	.20699	3.436	91.44	.20572	3.204	91.41	.20455
100	4.095	93.00	0.21104	3.774	92.96	0.20967	3.501	92.93	0.20841	3.266	92.91	0.20723
110	4.170	94.50	.21367	3.844	94.47	.21231	3.567	94.43	.21106	3.327	94.41	.20989
120	4.246	96.01	.21631	3.915	95.98	.21495	3.633	95.95	.21369	3.388	95.91	.21252
130	4.323	97.53	.21891	3.986	97.50	.21755	3.699	97.48	.21630	3.450	97.44	.21513
140	4.400	99.07	.22151	4.056	99.04	.22014	3.763	99.01	.21889	3.510	98.98	.21772
150	4.474	100.62	0.22406	4.126	100.59	0.22270	3.828	100.56	0.22144	3.571	100.53	0.22028
160	4.549	102.18	.22659	4.196	102.16	.22524	3.894	102.13	.22400	3.632	102.10	.22282
170	4.624	103.75	.22911	4.266	103.73	.22775	3.960	103.71	.22651	3.694	103.68	.22535
180	4.700	105.34	.23162	4.337	105.32	.23027	4.025	105.30	.22902	3.755	105.27	.22786
190	4.774	106.94	.23409	4.408	106.92	.23276	4.090	106.90	.23150	3.816	106.87	.23034
200	4.850	108.55	0.23656	4.478	108.55	0.23523	4.155	108.52	0.23398	3.877	108.49	0.23282
210	4.926	110.18	.23901	4.547	110.16	.23767	4.220	110.14	.23643	3.938	110.12	.23527
220	5.000	111.82	.24144	4.617	111.81	.24011	4.285	111.78	.23886	3.998	111.77	.23770
230	5.076	113.47	.24385	4.686	113.45	.24252	4.350	113.44	.24128	4.059	113.42	.24011
240	5.152	115.15	.24626	4.755	115.13	.24492	4.414	115.11	.24368	4.120	115.10	.24253
250	......	......	......	......	......	......	4.479	116.79	0.24607	4.181	116.78	0.24491

TABLE 12.—F12, DICHLORODIFLUOROMETHANE (CCl_2F_2).
PROPERTIES OF SUPERHEATED VAPOR.—(Continued.)

Temp. °F.	Abs. Pressure 16 lb./in.² Gage Pressure 1.3 lb./in.² (Sat'n. Temp.—18.0° F.)			Abs. Pressure 17 lb./in.² Gage Pressure 2.3 lb./in.² (Sat'n. Temp.—15.4° F.)			Abs. Pressure 18 lb./in.² Gage Pressure 3.3 lb./in.² (Sat'n. Temp.—12.9° F.)			Abs. Pressure 19 lb./in.² Gage Pressure 4.3 lb./in.² (Sat'n. Temp.—10.5° F.)		
t	V	H	S	V	H	S	V	H	S	V	H	S
(at sat'n)	(2.370)	(76.11)	(0.17254)	(2.240)	(76.41)	(0.17226)	(2.125)	(76.70)	(0.17202)	(2.012)	(76.99)	(0.17180)
−10	2.417	77.20	0.17498	2.268	77.15	0.17390	2.137	77.09	0.17289	2.022	77.05	0.17195
0	2.476	78.56	0.17800	2.325	78.52	0.17693	2.191	78.47	0.17592	2.072	78.42	0.17497
10	2.535	79.94	.18095	2.382	79.90	.17990	2.244	79.85	.17889	2.122	79.80	.17794
20	2.594	81.33	.18387	2.438	81.29	.18283	2.297	81.24	.18181	2.172	81.19	.18087
30	2.652	82.73	.18676	2.493	82.69	.18571	2.350	82.65	.18471	2.222	82.60	.18376
40	2.710	84.13	.18962	2.548	84.10	.18857	2.402	84.06	.18757	2.272	84.02	.18663
50	2.768	85.55	0.19244	2.603	85.52	0.19140	2.454	85.49	0.19041	2.321	85.45	0.18946
60	2.827	86.99	.19524	2.658	86.96	.19420	2.506	86.93	.19321	2.371	86.90	.19226
70	2.886	88.45	.19800	2.712	88.41	.19696	2.558	88.38	.19597	2.420	88.35	.19503
80	2.944	89.91	.20075	2.767	89.88	.19969	2.610	89.84	.19872	2.469	89.81	.19777
90	3.002	91.39	.20345	2.822	91.36	.20241	2.662	91.32	.20142	2.518	91.29	.20047
100	3.059	92.88	0.20613	2.876	92.85	0.20510	2.713	92.81	0.20411	2.567	92.78	0.20317
110	3.117	94.38	.20879	2.930	94.35	.20774	2.764	94.32	.20676	2.616	94.28	.20583
120	3.175	95.89	.21143	2.984	95.86	.21038	2.815	95.83	.20941	2.665	95.80	.20847
130	3.232	97.42	.21403	3.038	97.39	.21298	2.867	97.36	.21201	2.714	97.33	.21108
140	3.290	98.95	.21662	3.092	98.93	.21558	2.918	98.90	.21462	2.763	98.88	.21368
150	3.347	100.50	0.21917	3.147	100.48	0.21814	2.969	100.45	0.21717	2.812	100.43	0.21625
160	3.404	102.07	.22172	3.201	102.05	.22070	3.020	102.02	.21971	2.861	102.00	.21880
170	3.461	103.65	.22425	3.256	103.63	.22323	3.072	103.60	.22223	2.910	103.59	.22133
180	3.519	105.24	.22677	3.310	105.22	.22574	3.124	105.19	.22177	2.958	105.18	.22384
190	3.576	106.85	.22925	3.364	106.82	.22822	3.175	106.80	.22725	3.006	106.78	.22633
200	3.633	108.47	0.23173	3.417	108.44	0.23070	3.226	108.42	0.22973	3.054	108.39	0.22880
210	3.690	110.10	.23418	3.470	110.07	.23315	3.277	110.05	.23219	3.102	110.02	.23126
220	3.747	111.74	.23662	3.524	111.71	.23558	3.327	111.69	.23461	3.150	111.66	.23369
230	3.803	113.40	.23903	3.578	113.37	.23800	3.378	113.34	.23703	3.198	113.31	.23612
240	3.860	115.07	.24143	3.632	115.04	.24041	3.428	115.00	.23944	3.246	114.99	.23852
250	3.917	116.76	0.24382	3.686	116.72	0.24279	3.478	116.68	0.24182	3.294	116.68	0.24091
260	....	....	....	3.739	118.41	.24517	3.528	118.36	.24420	3.342	118.37	.24327

Temp. °F.	Abs. Pressure 20 lb./in.² Gage Pressure 5.3 lb./in.² (Sat'n. Temp.—8.2° F.)			Abs. Pressure 22 lb./in.² Gage Pressure 7.3 lb./in.² (Sat'n. Temp.—3.8° F.)			Abs. Pressure 24 lb./in.² Gage Pressure 9.3 lb./in.² (Sat'n. Temp. 0.3° F.)			Abs. Pressure 26 lb./in.² Gage Pressure 11.3 lb./in.² (Sat'n. Temp. 4.1° F.)		
(at sat'n)	(1.925)	(77.27)	(0.17160)	(1.762)	(77.77)	(0.17121)	(1.625)	(78.24)	(0.17089)	(1.510)	(78.68)	(0.17059)
0	1.965	78.39	0.17407	1.779	78.30	0.17237	....	....	....	....	....	....
10	2.013	79.76	.17704	1.823	79.68	.17534	1.664	79.59	0.17378	1.530	79.50	0.17233
20	2.060	81.14	.17996	1.866	81.07	.17827	1.705	80.99	.17673	1.568	80.90	.17529
30	2.107	82.55	.18286	1.909	82.48	.18118	1.745	82.40	.17964	1.606	82.31	.17821
40	2.155	83.97	.18573	1.953	83.90	.18405	1.785	83.82	.18251	1.643	83.74	.18109
50	2.203	85.40	0.18858	1.997	85.33	0.18691	1.825	85.26	0.18536	1.680	85.19	0.18395
60	2.250	86.85	.19138	2.040	86.78	.18971	1.865	86.71	.18817	1.716	86.64	.18675
70	2.297	88.31	.19415	2.083	88.25	.19248	1.905	88.17	.19096	1.754	88.10	.18954
80	2.343	89.78	.19688	2.125	89.71	.19522	1.944	89.64	.19371	1.790	89.57	.19229
90	2.390	91.26	.19959	2.168	91.19	.19794	1.984	91.13	.19643	1.826	91.05	.19501
100	2.437	92.75	0.20229	2.211	92.68	0.20064	2.023	92.63	0.19912	1.863	92.55	0.19772
110	2.483	94.26	.20494	2.253	94.19	.20330	2.062	94.13	.20179	1.900	94.06	.20039
120	2.530	95.78	.20759	2.296	95.71	.20595	2.101	95.64	.20444	1.936	95.58	.20305
130	2.577	97.31	.21020	2.339	97.25	.20856	2.140	97.17	.20704	1.972	97.12	.20565
140	2.623	98.85	.21280	2.381	98.80	.21116	2.179	98.72	.20966	2.008	98.67	.20827
150	2.669	100.40	0.21537	2.423	100.35	0.21373	2.218	100.28	0.21223	2.045	100.23	0.21085
160	2.716	101.97	.21792	2.465	101.92	.21628	2.257	101.86	.21479	2.082	101.80	.21341
170	2.762	103.56	.22045	2.507	103.50	.21881	2.296	103.45	.21732	2.117	103.39	.21594
180	2.808	105.15	.22297	2.550	105.10	.22133	2.335	105.05	.21984	2.153	104.99	.21847
190	2.854	106.76	.22545	2.592	106.71	.22382	2.373	106.66	.22233	2.189	106.60	.22096
200	2.901	108.38	0.22794	2.634	108.33	0.22630	2.412	108.28	0.22481	2.224	108.23	0.22344
210	2.947	110.01	.23039	2.676	109.96	.22876	2.450	109.92	.22727	2.259	109.86	.22589
220	2.992	111.65	.23283	2.717	111.60	.23120	2.489	111.57	.22972	2.295	111.51	.22834
230	3.038	113.31	.23524	2.758	113.26	.23361	2.528	113.23	.23214	2.331	113.17	.23076
240	3.084	114.98	.23766	2.800	114.94	.23603	2.566	114.90	.23455	2.366	114.85	.23318
250	3.130	116.67	0.24005	2.842	116.62	0.23843	2.605	116.58	0.23695	2.402	116.54	0.23557
260	3.177	118.36	.24242	2.884	118.32	.24080	2.644	118.28	.23933	2.438	118.24	.23796
270	3.223	120.07	.24477	2.926	120.04	.24316	2.682	120.00	.24169	2.474	119.95	.24033
280	....	....	....	....	....	....	....	....	....	2.510	121.68	.24267

TABLE 12.—F12, DICHLORODIFLUOROMETHANE (CCl_2F_2).
PROPERTIES OF SUPERHEATED VAPOR.—(Continued.)

Temp. °F.	Abs. Pressure 28 lb./in.², Gage Pressure 13.3 lb./in.² (Sat'n. Temp. 7.7° F.)			Abs. Pressure 30 lb./in.², Gage Pressure 15.3 lb./in.² (Sat'n. Temp. 11.1° F.)			Abs. Pressure 32 lb./in.², Gage Pressure 17.3 lb./in.² (Sat'n. Temp. 14.3° F.)			Abs. Pressure 34 lb./in.², Gage Pressure 19.3 lb./in.² (Sat'n. Temp. 17.4° F.)		
t	V	H	S	V	H	S	V	H	S	V	H	S
(at sat'n)	(1.409)	(79.10)	(0.17038)	(1.323)	(79.47)	(0.17008)	(1.245)	(79.84)	(0.16985)	(1.176)	(80.20)	(0.16965)
10	1.415	79.41	0.17099	.....	.....	.....	.....	.....	.....	.....	.....	.....
20	1.450	80.81	.17393	1.350	80.73	0.17269	1.262	80.67	0.17152	1.183	80.58	0.17040
30	1.485	82.23	.17685	1.383	82.15	.17562	1.293	82.08	.17445	1.212	82.00	.17333
40	1.520	83.66	.17975	1.415	83.58	.17851	1.323	83.51	.17734	1.241	83.44	.17623
50	1.555	85.11	0.18261	1.448	85.03	0.18138	1.354	84.96	0.18022	1.270	84.88	0.17910
60	1.590	86.56	.18544	1.480	86.48	.18420	1.384	86.41	.18304	1.299	86.34	.18194
70	1.625	88.03	.18823	1.512	87.95	.18699	1.414	87.88	.18583	1.328	87.81	.18474
80	1.659	89.51	.19097	1.544	89.43	.18974	1.444	89.36	.18860	1.356	89.29	.18750
90	1.693	90.99	.19371	1.576	90.91	.19249	1.474	90.85	.19133	1.385	90.78	.19025
100	1.727	92.49	0.19642	1.608	92.41	0.19519	1.504	92.35	0.19404	1.413	92.29	0.19295
110	1.761	94.01	.19909	1.640	93.93	.19787	1.535	93.87	.19673	1.441	93.81	.19563
120	1.795	95.53	.20174	1.672	95.46	.20053	1.565	95.40	.19940	1.470	95.34	.19831
130	1.828	97.07	.20436	1.703	97.00	.20315	1.595	96.94	.20202	1.498	96.88	.20094
140	1.862	98.62	.20698	1.735	98.54	.20577	1.624	98.50	.20463	1.526	98.43	.20356
150	1.896	100.18	0.20956	1.767	100.11	0.20836	1.654	100.06	0.20721	1.554	100.00	0.20614
160	1.930	101.75	.21212	1.799	101.69	.21092	1.683	101.64	.20977	1.582	101.58	.20871
170	1.963	103.33	.21466	1.829	103.28	.21344	1.713	103.23	.21232	1.610	103.17	.21125
180	1.997	104.93	.21719	1.860	104.88	.21597	1.743	104.83	.21486	1.638	104.78	.21379
190	2.030	106.55	.21967	1.891	106.49	.21846	1.772	106.45	.21735	1.666	106.40	.21629
200	2.063	108.17	0.22216	1.923	108.12	0.22096	1.802	108.08	0.21985	1.693	108.03	0.21878
210	2.096	109.81	.22462	1.954	109.76	.22312	1.831	109.72	.22231	1.721	109.67	.22125
220	2.129	111.46	.22706	1.986	111.41	.22588	1.860	111.36	.22476	1.749	111.32	.22370
230	2.163	113.12	.22949	2.017	113.08	.22830	1.889	113.03	.22718	1.776	112.98	.22613
240	2.196	114.80	.23191	2.048	114.75	.23072	1.918	114.72	.22960	1.804	114.66	.22856
250	2.229	116.49	0.23430	2.079	116.44	0.23312	1.948	116.41	0.23200	1.833	116.35	0.23097
260	2.262	118.19	.23669	2.110	118.15	.23550	1.977	118.11	.23439	1.860	118.06	.24335
270	2.295	119.91	.23905	2.141	119.87	.23787	2.006	119.82	.23676	1.888	119.79	.23573
280	2.329	121.65	.24141	2.172	121.60	.24023	2.035	121.55	.23912	1.916	121.54	.23809
290	.....	.....	.....	.....	.....	.....	2.065	123.30	.24146	1.944	123.31	.24043

Temp. °F.	Abs. Pressure 36 lb./in.², Gage Pressure 21.3 lb./in.² (Sat'n. Temp. 20.4° F.)			Abs. Pressure 38 lb./in.², Gage Pressure 23.3 lb./in.² (Sat'n. Temp. 23.2° F.)			Abs. Pressure 40 lb./in.², Gage Pressure 25.3 lb./in.² (Sat'n. Temp. 25.9° F.)			Abs. Pressure 42 lb./in.², Gage Pressure 27.3 lb./in.² (Sat'n. Temp. 28.5° F.)		
t	V	H	S	V	H	S	V	H	S	V	H	S
(at sat'n)	(1.113)	(80.64)	(0.16947)	(1.058)	(80.88)	(0.16931)	(1.009)	(81.16)	(0.16914)	(0.963)	(81.44)	(0.16897)
30	1.140	81.90	0.17227	1.076	81.82	0.17126	1.019	81.76	0.17030	0.967	81.65	0.16939
40	1.168	83.35	.17518	1.103	83.27	.17418	1.044	83.20	.17322	0.991	83.10	.17231
50	1.196	84.81	0.17806	1.129	84.72	0.17706	1.070	84.65	0.17612	1.016	84.56	0.17521
60	1.223	86.27	.18089	1.156	86.19	.17991	1.095	86.11	.17896	1.040	86.03	.17806
70	1.250	87.74	.18369	1.182	87.67	.18272	1.120	87.60	.18178	1.063	87.51	.18086
80	1.278	89.22	.18647	1.208	89.16	.18551	1.144	89.09	.18455	1.087	89.00	.18365
90	1.305	90.71	.18921	1.234	90.66	.18826	1.169	90.58	.18731	1.110	90.50	.18640
100	1.332	92.22	0.19193	1.260	92.17	0.19096	1.194	92.09	0.19004	1.134	92.01	0.18913
110	1.359	93.75	.19462	1.285	93.69	.19365	1.218	93.62	.19272	1.158	93.54	.19184
120	1.386	95.28	.19729	1.310	95.22	.19631	1.242	95.15	.19538	1.181	95.09	.19451
130	1.412	96.82	.19991	1.336	96.76	.19895	1.267	96.70	.19803	1.204	96.64	.19714
140	1.439	98.37	.20254	1.361	98.32	.20157	1.291	98.26	.20066	1.227	98.20	.19979
150	1.465	99.93	0.20512	1.387	99.89	0.20416	1.315	99.83	0.20325	1.250	99.77	0.20237
160	1.492	101.51	.20770	1.412	101.47	.20673	1.340	101.42	.20583	1.274	101.36	.20496
170	1.518	103.11	.21024	1.437	103.07	.20929	1.364	103.02	.20838	1.297	102.96	.20751
180	1.545	104.72	.21278	1.462	104.67	.21183	1.388	104.63	.21092	1.320	104.57	.21005
190	1.571	106.34	.21528	1.487	106.29	.21433	1.412	106.25	.21343	1.343	106.19	.21256
200	1.597	107.97	0.21778	1.512	107.93	0.21681	1.435	107.88	0.21592	1.365	107.82	0.21505
210	1.623	109.61	.22024	1.537	109.57	.21928	1.459	109.52	.21840	1.388	109.47	.21754
220	1.650	111.27	.22270	1.562	111.22	.22176	1.482	111.17	.22085	1.411	111.12	.22000
230	1.676	112.94	.22513	1.587	112.89	.22419	1.506	112.84	.22329	1.434	112.80	.22244
240	1.702	114.62	.22756	1.612	114.58	.22662	1.530	114.52	.22572	1.457	114.49	.22486
250	1.728	116.31	0.22996	1.637	116.28	0.22903	1.554	116.21	0.22813	1.480	116.19	0.22728
260	1.754	118.02	.23235	1.662	117.99	.23142	1.577	117.92	.23052	1.502	117.90	.22967
270	1.780	119.74	.23472	1.687	119.71	.23379	1.601	119.65	.23289	1.524	119.62	.23204
280	1.807	121.47	.23708	1.712	121.45	.23616	1.625	121.40	.23526	1.547	121.36	.23441
290	1.833	123.22	.23942	1.737	123.20	.23850	1.649	123.15	.23760	1.570	123.11	.23675
300	.....	.....	.....	1.762	124.95	0.24083	1.673	124.92	0.23994	1.592	124.87	0.23909

TABLE 12.—F12, DICHLORODIFLUOROMETHANE (CCl_2F_2).
PROPERTIES OF SUPERHEATED VAPOR.—(Continued.)

Temp °F.	Abs. Pressure 44 lb./in.² Gage Pressure 29.3 lb./in.² (Sat'n. Temp. 31.0° F.)			Abs. Pressure 46 lb./in.² Gage Pressure 31.3 lb./in.² (Sat'n. Temp. 33.5° F.)			Abs. Pressure 48 lb./in.² Gage Pressure 33.13 lb./in.² (Sat'n. Temp. 35.8° F.)			Abs. Pressure 50 lb./in.² Gage Pressure 35.3 lb./in.² (Sat'n. Temp. 38.3° F.)		
t	V	H	S	V	H	S	V	H	S	V	H	S
(at sat'n)	(0.922)	(81.72)	(0.16882)	(0.885)	(82.00)	(0.16867)	(0.849)	(82.25)	(0.16855)	(0.817)	(82.52)	(0.16841)
40	0.943	83.03	0.17142	0.899	82.94	0.17057	0.858	82.85	0.16974	0.821	82.76	0.16895
50	0.966	84.48	0.17432	0.921	84.40	0.17347	0.880	84.32	0.17266	0.842	84.24	0.17187
60	0.989	85.96	.17717	0.943	85.88	.17633	0.902	85.80	.17554	0.863	85.72	.17475
70	1.012	87.45	.18000	0.965	87.37	.17916	0.923	87.29	.17837	0.884	87.22	.17760
80	1.035	88.94	.18279	0.988	88.88	.18198	0.944	88.79	.18117	0.904	88.72	.18040
90	1.058	90.44	.18556	1.010	90.39	.18474	0.965	90.30	.18394	0.924	90.23	.18317
100	1.080	91.95	0.18828	1.031	91.90	0.18746	0.986	91.82	0.18668	0.944	91.75	0.18591
110	1.103	93.48	.19099	1.053	93.43	.19016	1.007	93.35	.18939	0.964	93.29	.18862
120	1.125	95.02	.19367	1.074	94.96	.19285	1.028	94.89	.19208	0.984	94.83	.19132
130	1.147	96.57	.19630	1.096	96.51	.19551	1.048	96.44	.19472	1.004	96.39	.19397
140	1.170	98.14	.19895	1.117	98.08	.19814	1.069	98.01	.19737	1.024	97.96	.19662
150	1.192	99.72	0.20154	1.139	99.66	0.20075	1.089	99.59	0.19997	1.044	99.54	0.19923
160	1.214	101.31	.20412	1.160	101.25	.20333	1.110	101.18	.20256	1.064	101.14	.20182
170	1.236	102.91	.20667	1.181	102.85	.20588	1.130	102.79	.20513	1.084	102.75	.20439
180	1.258	104.52	.20922	1.202	104.46	.20843	1.150	104.40	.20766	1.103	104.36	.20694
190	1.280	106.14	.21173	1.223	106.09	.21094	1.170	106.02	.21017	1.123	105.98	.20946
200	1.302	107.78	0.21424	1.244	107.73	0.21344	1.191	107.66	0.21269	1.142	107.62	0.21196
210	1.324	109.42	.21672	1.265	109.38	.21592	1.211	109.31	.21517	1.162	109.28	.21444
220	1.346	111.08	.21918	1.286	111.04	.21839	1.231	110.98	.21763	1.181	110.95	.21691
230	1.367	112.75	.22161	1.307	112.71	.22083	1.251	112.66	.22007	1.200	112.62	.21935
240	1.389	114.44	.22405	1.327	114.39	.22326	1.271	114.35	.22251	1.220	114.31	.22179
250	1.411	116.14	0.22646	1.348	116.09	0.22567	1.291	116.05	0.22492	1.239	116.00	0.22419
260	1.432	117.85	.22885	1.369	117.81	.22806	1.311	117.77	.22731	1.258	117.71	.22660
270	1.454	119.57	.23123	1.390	119.54	.23044	1.331	119.49	.22970	1.277	119.44	.22898
280	1.475	121.31	.23359	1.410	121.27	.23281	1.351	121.23	.23207	1.296	121.18	.23134
290	1.496	123.06	.23592	1.431	123.02	.23515	1.370	122.98	.23440	1.314	122.93	.23367
300	1.518	124.82	.23826	1.452	124.79	0.23749	1.390	124.75	0.23674	1.332	124.69	0.23600
310	1.539	126.59	.24058	1.472	126.57	.23981	1.410	126.53	.23907	1.350	126.45	.23831

Temp. °F.	Abs. Pressure 52 lb./in.² Gage Pressure 37.3 lb./in.² (Sat'n. Temp. 40.4° F.)			Abs. Pressure 54 lb./in.² Gage Pressure 39.3 lb./in.² (Sat'n. Temp. 42.5° F.)			Abs. Pressure 56 lb./in.² Gage Pressure 41.3 lb./in.² (Sat'n. Temp. 44.6° F.)			Abs. Pressure 58 lb./in.² Gage Pressure 43.3 lb./in.² (Sat'n. Temp. 46.7° F.)		
(at sat'n)	(0.788)	(82.76)	(0.16831)	(0.759)	(82.98)	(0.16820)	(0.734)	(83.22)	(0.16810)	(0.710)	(83.44)	(0.16800)
50	0.808	84.17	0.17114	0.774	84.07	0.17036	0.744	83.99	0.16965	0.716	83.91	0.16896
60	0.827	85.65	.17400	0.794	85.56	.17326	0.763	85.48	.17255	0.734	85.40	.17185
70	0.847	87.14	.17684	0.814	87.06	.17612	0.782	86.98	.17541	0.752	86.90	.17471
80	0.867	88.64	.17966	0.833	88.57	.17894	0.801	88.49	.17824	0.770	88.41	.17753
90	0.886	90.15	.18244	0.852	90.08	.18172	0.819	90.01	.18102	0.788	89.93	.18033
100	0.906	91.68	0.18518	0.871	91.61	0.18446	0.837	91.54	0.18377	0.806	91.46	0.18309
110	0.925	93.22	.18789	0.890	93.16	.18718	0.856	93.08	.18651	0.824	93.00	.18583
120	0.945	94.77	.19059	0.908	94.71	.18989	0.874	94.63	.18921	0.842	94.55	.18854
130	0.964	96.33	.19325	0.927	96.27	.19255	0.892	96.19	.19188	0.860	96.11	.19122
140	0.983	97.90	.19590	0.945	97.84	.19520	0.910	97.77	.19453	0.877	97.68	.19387
150	1.002	99.48	0.19850	0.964	99.43	0.19782	0.928	99.36	0.19715	0.894	99.26	0.19648
160	1.021	101.07	.20109	0.982	101.03	.20043	0.946	100.96	.19975	0.912	100.86	.19908
170	1.040	102.68	.20365	1.001	102.64	.20299	0.964	102.57	.20232	0.929	102.47	.20166
180	1.059	104.30	.20621	1.019	104.25	.20554	0.981	104.19	.20487	0.946	104.09	.20423
190	1.078	105.93	.20873	1.037	105.88	.20806	0.999	105.83	.20739	0.963	105.72	.20676
200	1.097	107.58	0.21125	1.055	107.52	0.21057	1.016	107.48	0.20991	0.980	107.36	0.20927
210	1.116	109.23	.21374	1.073	109.17	.21305	1.034	109.14	.21241	0.997	109.02	.21177
220	1.134	110.89	.21620	1.091	110.84	.21553	1.051	110.81	.21487	1.014	110.70	.21424
230	1.153	112.56	.21865	1.109	112.51	.21797	1.068	112.49	.21731	1.031	112.39	.21669
240	1.172	114.26	.22110	1.127	114.20	.22042	1.086	114.18	.21977	1.048	114.10	.21914
250	1.190	115.96	0.22352	1.145	115.91	0.22283	1.103	115.88	0.22218	1.064	115.82	0.22155
260	1.208	117.67	.22591	1.163	117.63	.22523	1.120	117.59	.22458	1.081	117.55	.22398
270	1.227	119.40	.22829	1.181	119.37	.22762	1.138	119.31	.22698	1.098	119.29	.22635
280	1.245	121.14	.23065	1.199	121.11	.22999	1.155	121.05	.22934	1.114	121.04	.22871
290	1.263	122.90	.23299	1.216	122.86	.23233	1.172	122.80	.23169	1.130	122.80	.23105
300	1.281	124.66	0.23532	1.234	124.63	0.23467	1.189	124.57	0.23403	1.147	124.57	0.23340
310	1.298	126.42	.23763	1.251	126.40	.23699	1.206	126.36	.23635	1.164	126.35	.23574
320	......	......	......	......	......	......	1.223	128.17	.23867	1.180	128.14	.23806

TABLE 12.—F12, DICHLORODIFLUOROMETHANE (CCl_2F_2).
PROPERTIES OF SUPERHEATED VAPOR.—(Continued.)

Temp. °F.	Abs. Pressure 60 lb./in.² Gage Pressure 45.3 lb./in.² (Sat'n. Temp. 48.7° F.)			Abs. Pressure 70 lb./in.² Gage Pressure 55.3 lb./in.² (Sat'n. Temp. 57.9° F.)			Abs. Pressure 80 lb./in.² Gage Pressure 65.3 lb./in.² (Sat'n. Temp. 66.3° F.)			Abs. Pressure 90 lb./in.² Gage Pressure 75.3 lb./in.² (Sat'n. Temp. 73.9° F.)		
t	V	H	S	V	H	S	V	H	S	V	H	S
(at sat'n)	(0.682)	(83.66)	(0.16791)	(0.59[illegible])	(84.61)	(0.1676[illegible])	(0.581)	(85.45)	(0.16716)	(0.466)	(86.21)	(0.16686)
50	0.690	83.83	0.16829				......	......	......	......	......	......
60	0.708	85.33	.17120	0.597	84.94	1.6810	......	......	......	......	......	......
70	0.726	86.84	.17407	0.612	86.44	17097	0.526	86.01	0.16919	......	......	......
80	0.743	88.35	.17689	0.628	87.96	17382	0.540	87.56	.17108	0.473	87.18	0.16862
90	0.760	89.87	.17968	0.643	89.49	17665	0.554	89.12	.17394	0.486	88.74	.17149
100	0.778	91.41	0.18246	0.658	91.03	0.17943	0.568	90.68	0.17675	0.499	90.31	0.17433
110	0.795	92.96	.18519	0.673	92.59	18219	0.582	92.26	.17954	0.511	91.89	17713
120	0.812	94.51	.18789	0.689	94.16	18493	0.596	93.84	.18229	0.523	93.48	.17990
130	0.829	96.07	.19056	0.704	95.75	18763	0.609	95.43	.18500	0.535	95.08	.18262
140	0.846	97.65	.19323	0.719	97.34	19030	0.623	97.03	.18771	0.547	96.69	.18533
150	0.863	99.24	0.19585	0.733	98.94	0.19293	0.636	98.64	0.19035	0.559	98.31	0.18799
160	0.880	100.84	.19846	0.748	100.54	19555	0.649	100.26	19298	0.571	99.94	.19065
170	0.897	102.45	.20104	0.763	102.16	19814	0.662	101.88	.19558	0.584	101.58	.19327
180	0.913	104.07	.20360	0.777	103.80	20071	0.675	103.52	.19817	0.596	103.23	19588
190	0.930	105.71	.20613	0.793	105.45	20325	0.688	105.18	.20073	0.607	104.89	.19845
200	0.946	107.36	0.20865	0.806	107.10	0.20579	0.701	106.84	0.20328	0.619	106.56	0.20101
210	0.962	109.02	.21113	0.820	108.76	20829	0.714	108.51	.20580	0.630	108.24	20353
220	0.979	110.69	.21361	0.835	110.43	21079	0.726	110.19	.20828	0.642	109.93	.20603
230	0.995	112.37	.21607	0.849	112.13	21325	0.739	111.88	.21076	0.653	111.63	20852
210	1.012	114.06	.21853	0.863	113.83	21570	0.751	113.58	.21321	0.665	113.35	21100
250	1.028	115.77	0.22094	0.878	115.55	0.21815	0.764	115.30	0.21566	0.676	115.08	0.21345
260	1.044	117.49	.22334	0.892	117.28	22057	0.777	117.03	.21809	0.688	116.82	21589
270	1.060	119.23	.22573	0.906	119.02	22296	0.789	118.78	.22049	0.699	118.57	.21831
280	1.076	120.97	.22810	0.920	120.76	.22534	0.802	120.54	.22289	0.710	120.33	22070
290	1.092	122.73	.23045	0.934	122.52	22770	0.814	122.30	22525	0.721	122.10	.22306
300	1.108	124.50	0.23280	0.948	124.29	0.23006	0.826	124.08	0.22760	0.732	123.88	0.22542
310	1.124	126.28	.23513	0.961	126.07	23239	0.839	125.88	.22995	0.743	125.67	22776
320	1.140	128.07	.23745	0.975	127.88	23471	0.851	127.70	.23229	0.754	127.43	23008
330	......	......	......	0.989	129.70	23702	0.864	129.52	.23461	0.765	129.31	23240

Temp. °F.	Abs. Pressure 100 lb./in.² Gage Pressure 85.3 lb./in.² (Sat'n. Temp. 80.9° F.)			Abs. Pressure 110 lb./in.² Gage Pressure 95.3 lb./in.² (Sat'n. Temp. 87.3° F.)			Abs. Pressure 120 lb./in.² Gage Pressure 105.3 lb./in.² (Sat'n. Temp. 93.4° F.)			Abs. Pressure 130 lb./in.² Gage Pressure 115.3 lb./in.² (Sat'n. Temp. 99.1° F.)		
(at sat'n)	(0.419)	(86.89)	(0.16659)	(0.382)	(87.50)	(0.16635)	(0.350)	(88.05)	(0.16610)	(0.323)	(88.547)	(0.16588)
90	0.430	88.32	0.16926	0.385	87.91	0.16711	......	......	......	......	......	......
100	0.442	89.93	0.17210	0.396	89.51	0.17001	0.357	89.13	0.16803	0.324	88.69	0.16615
110	0.454	91.54	.17493	0.407	91.12	17287	0.367	90.75	.17090	0.333	90.33	16905
120	0.465	93.15	.17773	0.417	92.74	17568	0.377	92.38	.17374	0.343	91.98	.17193
130	0.477	94.76	.18049	0.428	94.37	17845	0.387	94.01	.17654	0.353	93.61	17476
140	0.488	96.37	.18321	0.438	96.01	18122	0.397	95.65	.17932	0.362	95.30	.17756
150	0.499	97.99	0.18590	0.449	97.66	0.18394	0.407	97.30	0.18207	0.371	96.97	0.18030
160	0.510	99.63	.18856	0.459	99.31	.18660	0.417	98.96	.18474	0.380	98.65	18302
170	0.521	101.28	.19120	0.469	100.97	.18924	0.426	100.63	.18743	0.389	100.34	.18571
180	0.531	102.94	.19381	0.479	102.64	.19187	0.436	102.31	.19011	0.398	102.04	.18839
190	0.542	104.61	.19638	0.489	104.32	.19447	0.445	104.00	.19271	0.407	103.74	.19102
200	0.553	106.29	0.19894	0.499	106.01	0.19706	0.454	105.70	0.19529	0.416	105.45	0.19362
210	0.563	107.98	.20148	0.509	107.71	.19962	0.463	107.41	.19785	0.424	107.16	.19620
220	0.574	109.68	.20401	0.519	109.42	.20216	0.472	109.13	.20041	0.433	108.89	.19877
230	0.585	111.39	.20650	0.528	111.14	.20464	0.482	110.86	.20294	0.442	110.62	.20130
240	0.595	113.11	.20899	0.538	112.87	.20712	0.491	112.60	.20545	0.450	112.36	.20382
250	0.606	114.84	0.21145	0.548	114.61	0.20959	0.500	114.35	0.20792	0.458	114.11	0.20629
260	0.616	116.58	.21389	0.557	116.36	.21205	0.508	116.11	.21035	0.467	115.87	.20876
270	0.626	118.33	.21631	0.567	118.12	.21448	0.517	117.88	.21279	0.475	117.64	.21122
280	0.636	120.10	.21870	0.576	119.89	21690	0.526	119.66	.21521	0.483	119.42	.21364
290	0.646	121.88	.22108	0.586	121.68	21930	0.534	121.45	.21760	0.492	121.21	.21605
300	0.657	123.67	0.22347	0.595	123.48	0.22167	0.543	123.25	0.22000	0.500	123.01	0.21846
310	0.667	125.47	.22583	0.605	125.29	.22405	0.552	125.07	.22238	0.508	124.82	.22084
320	0.677	127.28	.22817	0.614	127.11	.22639	0.560	126.90	.22472	0.516	126.64	.22320
330	0.687	129.10	.23050	0.623	128.94	.22872	0.569	128.74	.22707	0.524	128.47	.22554
340	0.697	130.94	.23251	0.632	130.78	.23103	0.578	130.50	.22940	0.531	130.31	.22786
350	0.707	132.80	0.23510	0.641	132.63	0.23333	0.586	132.45	0.23171	0.539	132.17	0.23016
360	0.718	134.68	.23738	0.651	134.50	.23562	0.595	134.32	.23400	0.547	134.05	.23246
370	......	......	......							0.555	135.91	.23475

TABLE 12.—F12, DICHLORODIFLUOROMETHANE (CCl_2F_2). PROPERTIES OF SUPERHEATED VAPOR.—(Concluded.)

Temp. °F	Abs. Press. 140 lb./in.² Gage Press. 125.3 lb./in.² (Sat'n. Temp. 104.5°F.)			Abs. Press. 160 lb./in.² Gage Press. 145.3 lb./in.² (Sat'n. Temp. 114.5°F.)			Abs. Press. 180 lb./in.² Gage Press. 165.3 lb./in.² (Sat'n. Temp. 123.7°F.)			Abs. Press. 200 lb./in.² Gage Press. 185.3 lb./in.² (Sat'n. Temp. 132.1°F.)			Abs. Press. 220 lb./in.² Gage Press. 205.3 lb./in.² (Sat'n. Temp. 139.9°F.)		
t	V	H	S	V	H	S	V	H	S	V	H	S	V	H	S
(at sat'n)	(0.298)	(88.99)	(0.16566)	(0.259)	(89.77)	(0.16528)	(0.228)	(90.38)	(0.16476)	(0.208)	(90.86)	(0.16434)	(0.181)	(91.80)	(0.16375)
100	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......
110	0.304	89.92	0.16725	......	......	......	......	......	......	......	......	......	......	......	......
120	0.314	91.60	.17021	0.264	90.68	0.16682	......	......	......	......	......	......	......	......	......
130	0.323	93.28	.17306	0.273	92.40	.16977	0.233	91.47	0.16665	......	......	......	......	......	......
140	0.332	94.96	.17590	0.282	94.12	.17269	0.241	93.23	.16964	0.208	92.30	0.16661	0.181	91.52	0.16378
150	0.341	96.65	0.17868	0.290	95.84	0.17553	0.249	94.99	0.17254	0.216	94.10	0.16966	0.188	93.32	0.16685
160	0.350	98.34	.18142	0.298	97.57	.17832	0.257	96.75	.17541	0.224	95.90	.17262	0.195	95.13	.16986
170	0.358	100.03	.18412	0.306	99.31	.18106	0.265	98.52	.17823	0.231	97.70	.17551	0.202	96.94	.17282
180	0.366	101.72	.18678	0.313	101.05	.18377	0.272	100.29	.18102	0.238	99.51	.17838	0.209	98.75	.17576
190	0.374	103.42	.18941	0.321	102.80	.18646	0.280	102.07	.18377	0.245	101.32	.18115	0.216	100.57	.17861
200	0.383	105.14	0.19205	0.329	104.55	0.18913	0.287	103.85	0.18648	0.252	103.13	0.18388	0.223	102.39	0.18142
210	0.391	106.86	.19466	0.336	106.31	.19175	0.294	105.63	.18912	0.258	104.94	.18659	0.229	104.22	.18420
220	0.399	108.59	.19724	0.344	108.07	.19435	0.301	107.42	.19174	0.265	106.76	.18927	0.236	106.05	.18694
230	0.407	110.33	.19976	0.351	109.83	.19693	0.307	109.21	.19433	0.272	108.58	.19192	0.242	107.89	.18962
240	0.415	112.09	.20229	0.358	111.60	.19949	0.314	111.01	.19693	0.278	110.40	.19455	0.248	109.74	.19226
250	0.423	113.85	0.20479	0.366	113.38	0.20203	0.321	112.81	0.19947	0.284	112.23	0.19713	0.254	111.59	0.19487
260	0.431	115.63	.20728	0.373	115.17	.20453	0.327	114.62	.20199	0.290	114.06	.19967	0.259	113.44	.19745
270	0.439	117.42	.20974	0.380	116.97	.20700	0.334	116.44	.20449	0.296	115.89	.20217	0.265	115.30	.20001
280	0.447	119.22	.21219	0.387	118.78	.20946	0.340	118.26	.20698	0.302	117.73	.20467	0.271	117.16	.20255
290	0.455	121.03	.21461	0.394	120.60	.21189	0.347	120.09	.20944	0.308	119.58	.20715	0.277	119.03	.20504
300	0.462	122.85	0.21701	0.401	122.43	0.21432	0.353	121.92	0.21187	0.314	121.44	0.20961	0.282	120.91	0.20753
310	0.470	124.67	.21939	0.408	124.27	.21672	0.359	123.76	.21428	0.320	123.31	.21204	0.288	122.80	.21002
320	0.477	126.50	.22174	0.414	126.12	.21909	0.365	125.61	.21665	0.326	125.19	.21445	0.293	124.70	.21244
330	0.485	128.33	.22411	0.421	127.98	.22145	0.371	127.47	.21904	0.331	127.08	.21685	0.299	126.60	.21485
340	0.492	130.17	.22646	0.428	129.85	.22381	0.377	129.34	.22140	0.337	128.98	.21923	0.304	128.51	.21724
350	0.500	132.02	0.22880	0.435	131.73	0.22616	0.383	131.23	0.22374	0.343	130.89	0.22159	0.309	130.42	0.21960
360	0.507	133.89	.23109	0.442	133.63	.22849	0.390	133.13	.22608	0.348	132.81	.22392	0.314	132.34	.22195
370	0.515	135.78	.23336	0.448	135.55	.23079	0.396	135.05	.22840	0.354	134.74	.22624	0.320	134.27	.22430
380	......	......	......	......	......	......	0.402	136.98	.23072	0.360	136.68	.22856	0.325	136.21	.22665
390	......	......	......	......	......	......	0.408	138.91	.23301	0.365	138.63	.23085	0.330	138.16	.22895
400	......	......	......	......	......	......	0.414	140.85	0.23529	0.370	140.59	0.23314	0.385	140.12	0.23124

TABLE 13.—FREON 11, TRICHLOROMONOFLUOROMETHANE (CCLF).
PROPERTIES OF SATURATED VAPOR.

Temp.	Pressure		Volume		Density		Heat Content from —40°			Entropy from —40°		Temp
°F. t	Abs. lb./in.² p	Gage lb./in.² p_d	Liquid ft.³/lb. v_f	Vapor ft.³/lb. v_g	Liquid lb./ft.³ $1/v_f$	Vapor lb./ft.³ $1/v_g$	Liquid Btu./lb. h_f	Latent Btu./lb. h	Vapor Btu./lb. h_g	Liquid Btu./lb.°F. s_f	Vapor Btu./lb.°F. s_g	°F. t
−40	0.7391	28.42*	0.00988	44.21	101.25	0.02262	0.00	87.48	87.48	0.0000	0.2085	−40
−38	0.7916	28.31*	.00989	41.47	101.10	.02411	0.39	87.33	87.72	.0009	.2081	−38
−36	0.8471	28.20*	.00991	38.93	100.96	.02569	0.79	87.17	87.96	.0019	.2076	−36
−34	0.9060	28.08*	.00992	36.57	100.81	.02735	1.18	87.02	88.20	.0028	.2072	−34
−32	0.9682	27.95*	.00993	34.37	100.66	.02910	1.58	86.86	88.44	.0037	.2068	−32
−30	1.034	27.81*	0.00995	32.33	100.52	0.03093	1.97	86.70	88.67	0.0046	0.2064	−30
−28	1.103	27.67*	.00996	30.44	100.37	.03285	2.36	86.55	88.91	.0055	.2060	−28
−26	1.176	27.53*	.00998	28.68	100.22	.03487	2.75	86.40	89.15	.0064	.2057	−26
−24	1.253	27.37*	.00999	27.03	100.07	.03700	3.15	86.24	89.39	.0073	.2053	−24
−22	1.334	27.20*	.01001	25.50	99.92	.03922	3.55	86.08	89.63	.0082	.2049	−22
−20	1.420	27.03*	0.01002	24.06	99.77	0.04157	3.94	85.93	89.87	0.0091	0.2046	−20
−18	1.510	26.85*	.01004	22.72	99.63	.04401	4.33	85.78	90.11	.0100	.2043	−18
−16	1.605	26.65*	.01005	21.47	99.48	.04658	4.73	85.62	90.35	.0109	.2040	−16
−14	1.705	26.45*	.01007	20.30	99.33	.04927	5.12	85.47	90.59	.0118	.2036	−14
−12	1.810	26.24*	.01008	19.20	99.18	.05209	5.52	85.31	90.83	.0127	.2033	−12
−10	1.920	26.01*	0.01010	18.17	99.03	0.05503	5.91	85.16	91.07	0.0136	0.2030	−10
− 8	2.035	25.78*	.01011	17.21	98.87	.05810	6.31	85.00	91.31	.0145	.2027	− 8
− 6	2.156	25.53*	.01013	16.32	98.72	.06129	6.70	84.85	91.55	.0153	.2024	− 6
− 4	2.283	25.27*	.01015	15.47	98.57	.06464	7.10	84.69	91.79	.0162	.2021	− 4
− 2	2.416	25.00*	.01016	14.68	98.42	.06813	7.49	84.54	92.03	.0171	.2018	− 2
0	2.555	24.72*	0.01018	13.94	98.27	0.07176	7.89	84.38	92.27	0.0179	0.2015	0
2	2.700	24.42*	.01019	13.24	98.11	.07554	8.28	84.23	92.51	.0188	.2013	2
4	2.852	24.11*	.01021	12.58	97.96	.07949	8.68	84.07	92.75	.0197	.2010	4
5†	2.931	23.95*	.01022	12.27	97.88	.08152	8.88	84.00	92.88	.0201	.2009	5†
6	3.012	23.79*	.01022	11.96	97.81	.08361	9.08	83.92	93.00	.0205	.2008	6
8	3.179	23.45	.01024	11.38	97.65	.08790	9.48	83.76	93.24	.0213	.2005	8
10	3.352	23.10*	0.01026	10.83	97.50	0.09233	9.88	83.60	93.48	0.0222	0.2003	10
12	3.534	22.73*	.01027	10.31	97.34	.09697	10.28	83.45	93.72	.0231	.2000	12
14	3.724	22.34*	.01029	9.823	97.19	.1018	10.68	83.29	93.97	.0239	.1998	14
16	3.923	21.94*	.01031	9.359	97.03	.1068	11.07	83.14	94.21	.0248	.1996	16
18	4.129	21.52*	.01032	8.925	96.88	.1120	11.47	82.98	94.45	.0256	.1993	18
20	4 342	21.08*	0.01034	8.519	96.72	0.1174	11.87	82.82	94.69	0.0264	0.1991	20
22	4.567	20.62*	.01036	8.129	96.57	.1230	12.27	82.66	94.94	.0273	.1989	22
24	4.801	20.15*	.01037	7.760	96.41	.1289	12.68	82.50	95.18	.0281	.1987	24
26	5.043	19.66*	.01039	7.414	96.25	.1349	13.08	82.34	95.42	.0289	.1985	26
28	5.294	19.14*	.01041	7.087	96.10	.1411	13.48	82.18	95.66	.0297	.1983	28
30	5.557	18.61*	0.01042	6.776	95.94	0.1476	13.88	82.03	95.91	0.0306	0.1981	30
32	5.830	18.05*	.01044	6.481	95.78	.1543	14.28	81.87	96.15	.0314	.1979	32
34	6 115	17.47*	.01046	6.200	95.62	.1613	14.68	81.71	96.39	.0322	.1977	34
36	6 411	16.87*	.01048	5.934	95.46	.1685	15.08	81.55	96.63	.0330	.1976	36
38	6.718	16.25*	.01049	5.682	95.30	.1760	15.49	81.38	96.87	.0338	.1974	38
40	7.032	15.61*	0.01051	5.447	95.14	0.1836	15.89	81.22	97.11	0.0346	0.1972	40
42	7.362	14.94*	.01053	5.220	94.98	.1916	16.30	81.06	97.36	.0354	.1970	42
44	7.702	14.24*	.01055	5.006	94.82	.1998	16.70	80.90	97.60	.0362	.1969	44
46	8.055	13.52*	.01056	4.802	94.66	.2083	17.11	80.73	97.84	.0370	.1967	46
48	8.422	12.78*	.01058	4.607	94.50	.2170	17.52	80.57	98.08	.0378	.1966	48
50	8.804	12.00*	0.01060	4.421	94.34	0.2262	17.92	80.40	98.32	0.0386	0.1964	50
52	9 199	11.20*	.01062	4 245	94.18	.2356	18.33	80.24	98 56	.0394	.1963	52
54	9.605	10.37*	.01064	4.078	94.02	.2452	18.74	80.07	98.81	.0402	.1961	54
56	10.02	9.53*	.01066	3.921	93.85	.2550	19.15	79.90	99.05	.0410	.1960	56
58	10.45	8.65*	.01067	3.770	93.69	.2652	19.56	79.73	99.29	.0418	.1959	58
60	10.90	7.73*	0.01069	3.626	93.53	0.2758	19.96	79.57	99.53	0.0426	0.1958	60
62	11 37	6.78*	.01071	3.487	93.36	.2868	20.37	79.40	99.77	.0434	.1956	62
64	11.85	5.80*	.01073	3.356	93.20	.2980	20.78	79.23	100.01	.0442	.1955	64
66	12.35	4.78*	.01075	3.229	93.04	.3097	21.19	79.06	100.25	.0450	.1954	66
68	12.87	3.72*	.01077	3.107	92.87	.3219	21.61	78.88	100.49	.0457	.1953	68
70	13.40	2.64*	0.01079	2.993	92.71	0.3342	22.02	78.71	100.73	0.0465	0.1951	70
72	13.95	1.53*	.01081	2.883	92.54	.3469	22.43	78.54	100.97	.0473	.1950	72
74	14.51	0.39*	.01083	2.779	92.38	.3598	22.84	78.37	101.21	.0481	.1949	74
76	15.09	0.39	.01085	2.679	92.21	.3732	23.26	78.19	101.45	.0489	.1948	76
78	15.69	0.99	.01086	2.584	92.04	.3870	23.68	78.01	101.69	.0496	.1947	78

* Inches of mercury below one atmosphere
† Standard ton temperature

TABLE 13.—FREON 11, TRICHLOROMONOFLUOROMETHANE (CCl_3F). PROPERTIES OF SATURATED VAPOR.—(Concluded.)

Temp.	Pressure		Volume		Density		Heat Content from −40°			Entropy from −40°		Temp.
°F. t	Abs. lb./in.² p	Gage lb./in.² p_d	Liquid ft.³/lb. v_f	Vapor ft.³/lb. v_g	Liquid lb./ft.³ $1/v_f$	Vapor lb./ft.³ $1/v_g$	Liquid Btu/lb. h_f	Latent Btu/lb. h	Vapor Btu/lb. h_g	Liquid Btu/lb. °F s_f	Vapor Btu/lb. °F s_g	°F t
80	16.31	1.61	0.01088	2.492	91.88	0.4012	24.09	77.84	101.93	0.0504	0.1947	80
82	16.94	2.24	.01090	2.406	91.71	.4157	24.51	77.66	102.17	.0512	.1946	82
84	17.60	2.90	.01092	2.322	91.54	.4307	24.93	77.48	102.41	.0519	.1945	84
86†	18.28	3.58	.01094	2.242	91.38	.4461	25.34	77.31	102.65	.0527	.1944	86†
88	18.97	4.27	.01096	2.165	91.21	.4619	25.76	77.13	102.89	.0535	.1943	88
90	19.69	4.99	0.01098	2.091	91.04	0.4783	26.18	76.95	103.12	0.0542	0.1942	90
92	20.43	5.73	.01101	2.020	90.87	.4950	26.60	76.76	103.36	.0550	.1941	92
94	21.19	6.49	.01103	1.952	90.70	.5122	27.01	76.58	103.59	.0557	.1941	94
96	21.97	7.27	.01105	1.887	90.53	.5299	27.43	76.40	103.83	.0565	.1940	96
98	22.77	8.07	.01107	1.825	90.36	.5480	27.85	76.21	104.07	.0572	.1939	98
100	23.60	8.90	0.01109	1.765	90.19	0.5666	28.27	76.03	104.30	0.0580	0.1938	100
102	24.45	9.75	.01111	1.707	90.02	.5857	28.70	75.84	104.54	.0587	.1938	102
104	25.33	10.63	.01113	1.652	89.85	.6054	29.12	75.65	104.77	.0595	.1937	104
106	26.23	11.53	.01115	1.599	89.68	.6256	29.54	75.46	105.00	.0602	.1937	106
108	27.15	12.45	.01117	1.548	89.51	.6461	29.97	75.27	105.24	.0610	.1936	108
110	28.09	13.39	0.01119	1.499	89.34	0.6671	30.40	75.08	105.47	0.0617	0.1935	110
112	29.05	14.35	.01122	1.452	89.16	.6885	30.82	74.89	105.71	.0625	.1935	112
114	30.04	15.34	.01124	1.407	88.99	.7107	31.24	74.70	105.9[illegible]	.0632	.1935	114
116	31.07	16.37	.01126	1.363	88.82	.7335	31.67	74.50	106.17	.0639	.1934	116
118	32.11	17.41	.01128	1.321	88.65	.7570	32.10	74.30	106.40	.0647	.1933	118
120	33.20	18.50	0.01130	1.281	88.47	0.7808	32.53	74.10	106.63	0.0654	0.1933	120
122	34.29	19.59	.01133	1.243	88.30	.8049	32.95	73.91	106.86	.0661	.1932	122
124	35.42	20.72	.01135	1.206	88.12	.8296	33.38	73.71	107.09	.0669	.1932	124
126	36.56	21.86	.01137	1.170	87.95	.8550	33.81	73.51	107.32	.0676	.1931	126
128	37.74	23.04	.01139	1.135	87.77	.8811	34.24	73.31	107.55	.0683	.1931	128
130	38.96	24.26	0.01142	1.101	87.60	0.9080	34.67	73.11	107.78	0.0691	0.1931	130
132	40.23	25.53	.01144	1.068	87.42	.9361	35.10	72.90	108.00	.0698	.1930	132
134	41.50	26.80	.01146	1.037	87.25	.9646	35.54	72.69	108.23	.0705	.1929	134
136	42.80	28.10	.01149	1.007	87.07	0.9927	35.97	72.49	108.46	.0712	.1929	136
138	44.12	29.42	.01151	0.9785	86.88	1.022	36.40	72.28	108.68	.0719	.1929	138
140	45.50	30.80	0.01154	0.9505	86.69	1.052	36.84	72.07	108.91	0.0727	0.1929	140
142	46.92	32.22	.01156	.9231	86.50	1.083	37.28	71.85	109.13	.0734	.1928	142
144	48.35	33.65	.01159	.8970	86.32	1.115	37.71	71.64	109.35	.0741	.1928	144
146	49.81	35.11	.01161	.8719	86.14	1.147	38.15	71.43	109.58	.0748	.1928	146
148	51.31	36.61	.01163	.8476	85.96	1.180	38.59	71.21	109.80	.0755	.1927	148
150	52.85	38.15	0.01166	0.8240	85.78	1.214	39.02	71.00	110.02	0.0763	0.1927	150
152	54.41	39.71	.01168	.8014	85.60	1.248	39.46	70.78	110.24	.0770	.1927	152
154	56.01	41.31	.01171	.7794	85.41	1.283	39.91	70.56	110.47	.0777	.1927	154
156	57.65	42.95	.01173	.7581	85.23	1.319	40.35	70.34	110.69	.0784	.1927	156
158	59.32	44.62	.01176	.7376	85.04	1.356	40.79	70.12	110.90	.0791	.1926	158
160	61.04	46.34	0.01179	0.7176	84.85	1.394	41.23	69.89	111.12	0.0798	0.1926	160

† Standard ton temperature.

TABLE 14.—PROPERTIES OF "FREON-22" ($CHClF_2$) (LIQUID AND SATURATED VAPOR).

Temp. °F t	Pressure #/sq. in.		Volume cu. ft./lb.		Density #/cu. ft.		Heat Content Btu./#			Entropy Btu./# °F	
	abs. p	gauge	Liquid v_f	Vapor v_g	Liquid $1/v_f$	Vapor $1/v_g$	Liquid h_f	Latent h	Vapor h_g	Liquid s_f	Vapor s_g
−155	.199	*29.51	.0102	188.13	97.67	.0053156	−29.05	115.85	86.80	−0.08075	0.29958
−150	.260	*29.39	.0103	146.06	97.33	.0068467	−27.77	115.15	87.38	− .07670	.29523
−145	.338	*29.23	.0103	114.51	96.99	.0087329	−26.50	114.46	87.96	− .07265	.29118
−140	.433	*29.04	.0103	90.613	96.63	.011036	−25.23	113.78	88.55	− .06865	.28736
−135	.551	*28.80	.0104	72.465	96.27	.013799	−23.98	113.10	89.12	− .06471	.28375
−130	.695	*28.51	.0104	58.214	95.91	.017178	−22.71	112.43	89.72	− .06085	.28026
−125	.869	*28.15	.0105	47.226	95.53	.021175	−21.45	111.76	90.31	− .05706	.27695
−120	1.079	*27.72	.0105	38.600	95.15	.025907	−20.20	111.10	90.90	− .05335	.27380
−115	1.329	*27.21	.0106	31.773	94.76	.031473	−18.96	110.45	91.49	− .04970	.27082
−110	1.626	*26.61	.0106	26.329	94.37	.037981	−17.71	109.80	92.09	− .04609	.26798
−105	1.976	*25.90	.0106	21.960	93.97	.045538	−16.46	109.15	92.69	− .04254	.26527
−100	2.386	*25.06	.0107	18.426	93.56	.054272	−15.21	108.50	93.29	− .03903	.26269
− 95	2.865	*24.09	.0107	15.544	93.14	.064333	−13.96	107.85	93.89	− .03557	.26023
− 90	3.417	*22.96	.0108	13.196	92.72	.075783	−12.71	107.20	94.49	− .03216	.25788
− 85	4.055	*21.67	.0108	11.256	92.29	.088843	−11.45	106.55	95.10	− .02881	.25563
− 80	4.787	*20.18	.0109	9.6497	91.85	.10363	−10.20	105.90	95.70	− .02551	.25347
− 78	5.13	*19.48	.0109	9.0510	91.67	.11049	− 9.69	105.63	95.94	− .02419	.25262
− 76	5.48	*18.77	.0109	8.5045	91.49	.11758	− 9.19	105.37	96.18	− .02288	.25179
− 74	5.83	*18.05	.0109	8.0115	91.31	.12482	− 8.68	105.10	96.42	− .02158	.25098
− 72	6.20	*17.31	.0110	7.5641	91.13	.13220	− 8.18	104.82	96.66	− .02028	.25020
− 70	6.57	*16.55	.0110	7.1555	90.95	.13975	− 7.67	104.57	96.90	− .01899	.24943
− 68	6.97	*15.74	.0110	6.7806	90.77	.14747	− 7.16	104.30	97.14	− .01770	.24868
− 66	7.40	*14.87	.0110	6.4178	90.58	.15581	− 6.66	104.04	97.38	− .01641	.24795
− 64	7.84	*13.96	.0111	6.0750	90.39	.16460	− 6.15	103.81	97.62	− .01512	.24722
− 62	8.33	*12.97	.0111	5.7526	90.21	.17383	− 5.65	103.51	97.86	− .01384	.24650
− 60	8.86	*11.89	.0111	5.4520	90.03	.18342	− 5.14	103.24	98.10	− .01256	.24580
− 58	9.41	*10.78	.0111	5.1701	89.84	.19342	− 4.62	102.96	98.34	− .01129	.24512

−56	9.98	* 9.62	.0112	4.9039	89.65	.20392	−4.11	102.69	98.58	−.01002	.24445
−54	10.56	* 8.44	.0112	4.6536	89.46	.21489	−3.59	102.41	98.82	−.00875	.24378
−52	11.15	* 7.24	.0112	4.4172	89.27	.22639	−3.08	102.14	99.06	−.00749	.24311
−50	11.74	* 6.02	.0112	4.1948	89.08	.23839	−2.56	101.86	99.30	−.00623	.24245
−48	12.39	* 4.70	.0112	3.9872	88.88	.25080	−2.04	101.58	99.54	−.00498	.24181
−46	13.07	* 3.33	.0113	3.7936	88.68	.26360	−1.52	101.30	99.78	−.00373	.24119
−44	13.78	* 1.38	.0113	3.6114	88.49	.27690	−1.00	101.02	100.02	−.00248	.24059
−42	14.52	* 0.36	.0113	3.4399	88.30	.29070	−.48	100.74	100.26	−.00124	.24000
−40	15.31	0.609	.0114	3.2787	88.10	.30500	0.04	100.46	100.50	.0	.23942
−38	16.16	1.459	.0114	3.1250	87.90	.32000	.57	100.17	100.74	.00124	.23884
−36	17.02	2.319	.0114	2.9828	87.70	.33525	1.11	99.86	100.97	.00247	.23827
−34	17.90	3.199	.0114	2.8491	87.50	.35100	1.66	99.54	101.20	.00370	.23771
−32	18.79	4.089	.0115	2.7223	87.29	.36733	2.21	99.22	101.43	.00493	.23717
−30	19.69	4.989	.0115	2.5934	87.09	.38559	2.75	98.91	101.66	.00616	.23663
−28	20.71	6.014	.0115	2.4712	86.89	.40466	3.30	98.59	101.89	.00730	.23609
−26	21.76	7.064	.0116	2.3594	86.69	.42383	3.86	98.26	102.12	.00844	.23556
−24	22.83	8.125	.0116	2.2568	86.48	.44310	4.40	97.95	102.35	.00958	.23505
−22	23.90	9.200	.0116	2.1622	86.27	.46249	4.96	97.62	102.58	.01072	.23455
−20	24.99	10.292	.0116	2.0748	86.06	.48197	5.51	97.30	102.81	.01186	.23406
−18	26.16	11.462	.0117	1.9854	85.85	.50367	6.07	96.97	103.04	.01300	.23358
−16	27.38	12.682	.0117	1.9016	85.64	.52587	6.61	96.65	103.26	.01414	.23311
−14	28.65	13.953	.0117	1.8227	85.43	.54863	7.17	96.32	103.49	.01530	.23264
−12	29.97	15.273	.0117	1.7488	85.21	.57191	7.67	96.04	103.71	.01643	.23217
−10	31.34	16.644	.0118	1.6781	84.99	.59591	8.21	95.67	103.88	.01757	.23168
−8	32.78	18.084	.0118	1.6086	84.78	.62165	8.71	95.33	104.04	.01871	.23120
−6	34.26	19.557	.0118	1.5434	84.56	.64792	9.20	95.01	104.21	.01985	.23074
−4	35.76	21.057	.0119	1.4821	84.34	.67471	9.70	94.67	104.37	.02099	.23029
−2	37.30	22.597	.0119	1.4245	84.12	.70201	10.19	94.35	104.54	.02213	.22985
0	38.87	24.170	.0119	1.3702	83.90	.72980	10.74	94.01	104.75	.02327	.22941
2	40.55	25.85	.0119	1.3193	83.68	.7580	11.28	93.68	104.96	.02442	.22896
4	42.25	27.55	.0120	1.2690	83.46	.7880	11.82	93.36	105.18	.02556	.22852
5†	**43.12**	**28.42**	**.0120**	**1.2421**	**83.34**	**.8051**	**12.09**	**93.20**	**105.29**	**.02613**	**.22830**
6	44.01	29.31	.0120	1.2195	83.12	.8200	12.36	93.03	105.39	.02670	.22807

*Inches mercury below one atmosphere.
†Standard ton temperature

TABLE 14.—PROPERTIES OF "FREON-22" ($CHClF_2$) (LIQUID AND SATURATED VAPOR)—(Concluded.)

Temp. °F t	Pressure #/sq. in. abs. p	Pressure #/sq. in. gauge	Volume cu. ft./lb. Liquid v_f	Volume cu. ft./lb. Vapor v_g	Density #/cu. ft. Liquid $1/v_f$	Density #/cu. ft. Vapor $1/v_g$	Heat Content Btu./# Liquid h_f	Heat Content Btu./# Latent h	Heat Content Btu./# Vapor h_g	Entropy Btu./# °F Liquid s_f	Entropy Btu./# °F Vapor s_g
8	45.84	31.14	.0121	1.1733	82.90	.8523	12.90	92.71	105.61	.02784	.22763
10	47.66	32.96	.01208	1.1295	82.78	.88532	13.44	92.38	105.82	.02898	.22720
12	49.65	34.95	.01212	1.0872	82.55	.9198	13.98	92.05	106.03	.03012	.22677
14	51.68	36.98	.01215	1.0473	82.32	.9548	14.52	91.72	106.24	.03126	.22635
16	53.74	39.04	.01218	1.0080	82.09	.9920	15.06	91.40	106.46	.03240	.22593
18	55.85	41.15	.01222	0.97087	81.86	1.0300	15.60	91.07	106.67	.03354	.22551
20	58.00	43.30	.01225	0.93555	81.63	1.0688	16.16	90.73	106.89	.03468	.22509
22	60.30	45.60	.01229	.90278	81.39	1.1076	16.70	90.41	107.11	.03585	.22468
24	62.64	47.94	.01233	.87053	81.16	1.1487	17.24	90.09	107.33	.03702	.22428
26	65.03	50.33	.01237	.83899	80.92	1.1919	17.80	89.76	107.56	.03820	.22389
28	67.37	52.77	.01239	.80997	80.69	1.2346	18.33	89.43	107.76	.03937	.22350
30	69.97	55.27	.01243	.78094	80.45	1.2805	18.85	89.10	107.95	.04054	.22314
32	72.52	57.82	.01247	.75283	80.21	1.3283	19.38	88.77	108.15	.04172	.22278
34	75.19	60.47	.01251	.72640	79.97	1.3766	19.91	88.43	108.34	.04289	.22242
36	77.92	63.22	.01254	.70153	79.93	1.4254	20.44	88.09	108.53	.04407	.22206
38	80.77	66.07	.01258	.67809	79.49	1.4747	20.98	87.74	108.72	.04524	.22170
40	83.72	69.02	.01262	.65591	79.25	1.5246	21.52	87.39	108.91	.04642	.22134
42	86.77	72.07	.01266	.63339	79.00	1.5788	22.12	86.97	109.09	.04761	.22100
44	89.85	75.15	.01270	.61199	78.76	1.6340	22.72	86.55	109.27	.04879	.22066
46	92.97	78.27	.01274	.59161	78.51	1.6903	23.33	86.12	109.45	.04998	.22032
48	96.15	81.45	.01277	.57221	78.27	1.7476	23.94	85.69	109.63	.05116	.21998
50	99.40	84.70	.01282	.55371	78.02	1.8060	24.55	85.25	109.80	.05235	.21964
52	102.76	88.06	.01286	.53547	77.77	1.8675	25.17	84.79	109.96	.05355	.21931
54	106.22	91.52	.01290	.51794	77.51	1.9307	25.79	84.33	110.12	.05475	.21897
56	109.78	95.08	.01293	.50112	77.26	1.9955	26.41	83.87	110.28	.05594	.21864
58	113.44	98.74	.01298	.48498	77.01	2.0619	27.03	83.41	110.44	.05713	.21830

60	117.2	102.5	.01303	.46951	76.75	2.1299	27.65	82.95	110.60	.05833	.21797
60	121.14	106.34	.01307	.45435	76.49	2.2009	28.28	82.46	110.74	.05954	.21765
64	124.95	110.25	.01312	.43987	76.23	2.2734	28.91	81.97	110.88	.06074	.21733
66	128.94	114.24	.01316	.42600	75.98	2.3474	29.54	81.48	111.02	.06195	.21700
68	133.02	118.32	.01321	.41272	75.72	2.4229	30.17	80.99	111.16	.06315	.21669
70	137.2	122.5	.01325	.40000	75.46	2.5000	30.81	80.50	111.31	.06436	.21636
72	141.5	126.8	.01330	.38734	75.20	2.5817	31.47	79.97	111.44	.06556	.21603
74	145.9	131.2	.01335	.37519	74.94	2.6653	32.12	79.45	111.57	.06676	.21570
76	150.4	135.7	.01339	.36354	74.68	2.7507	32.78	78.92	111.70	.06796	.21537
78	155.0	140.3	.01344	.35239	74.42	2.8377	33.43	78.40	111.83	.06916	.21503
80	159.7	145.0	.01349	.34174	74.15	2.9262	34.09	77.86	111.95	.07035	.21469
82	164.5	149.8	.01353	.33146	73.88	3.0169	34.76	77.30	112.06	.07154	.21432
84	169.4	154.7	.01358	.32143	73.61	3.1111	35.43	76.74	112.17	.07273	.21394
86†	**174.4**	**159.7**	**.01363**	**.31165**	**73.34**	**3.2087**	**36.09**	**76.19**	**112.28**	**.07392**	**.21356**
88	179.5	164.8	.01368	.30212	73.07	3.3099	36.76	75.63	112.39	.07511	.21318
90	184.8	170.1	.01374	.29284	72.81	3.4148	37.43	75.06	112.49	.07630	.21281
92	190.1	175.4	.01380	.28407	72.52	3.5202	38.10	74.47	112.57	.07749	.21245
94	195.5	180.8	.01384	.27560	72.23	3.6285	38.78	73.87	112.65	.07867	.21209
96	201.1	186.4	.01389	.26733	71.94	3.7406	39.45	73.28	112.73	.07985	.21173
98	206.8	192.1	.01395	.25936	71.65	3.8556	40.13	72.68	112.81	.08103	.21137
100	212.6	197.9	.01402	.25169	71.35	3.9731	40.80	72.08	112.88	.08221	.21102
102	218.5	203.8	.01408	.24429	71.04	4.0935	41.47	71.45	112.92	.08338	.21064
104	224.5	209.8	.01414	.23710	70.73	4.2176	42.15	70.81	112.96	.08456	.21026
106	230.7	216.0	.01420	.23011	70.42	4.3457	42.82	70.19	113.01	.08574	.20988
108	237.0	222.3	.01426	.22332	70.10	4.4778	43.50	69.56	113.06	.08692	.20950
110	243.4	228.7	.01433	.21673	69.78	4.6140	44.17	68.94	113.11	.08810	.20913
112	249.9	235.2	.01440	.21057	69.45	4.7490	44.87	68.28	113.15	.08928	.20876
114	256.6	241.9	.01447	.20453	69.12	4.8893	45.57	67.62	113.19	.09046	.20839
116	263.4	248.7	.01454	.19861	68.78	5.0349	46.27	66.97	113.24	.09164	.20802
118	270.3	255.6	.01461	.19280	68.44	5.1867	46.97	66.32	113.29	.09281	.20765
120	277.3	262.6	.01469	.18709	68.10	5.3451	47.67	65.67	113.34	.09398	.20728

*Inches mercury below one atmosphere.
†Standard ton temperature

TABLE 15.—SATURATED CARBON DIOXIDE (CO_2).

Temp.	Pressure	Volume		Total heat from −40°			Entropy from 32° plus 1	
°F. t	Abs., lb./in.² p	Liquid, ft.³/lb. v_f	Vapor, ft.³/lb. v_g	Liquid, Btu./lb. h_f	Latent, Btu./lb. h	Vapor, Btu./lb. h_g	Liquid, Btu./lb.°F. s_f	Vapor, Btu./lb.°F. s_g
−10	257.3	0.01532	0.3472	13.9	124.8	138.7	.9532	1.2303
−8	266.5	0.01540	0.3349	14.9	123.9	138.8	.9552	1.2297
−6	275.9	0.01547	0.3231	15.9	122.9	138.8	.9573	1.2284
−4	285.4	0.01555	0.3118	16.9	122.0	138.9	.9594	1.2272
−2	295.3	0.01563	0.3009	17.9	121.0	138.9	.9614	1.2259
0	305.5	0.01570	0.2904	18.8	120.1	138.9	.9636	1.2247
2	315.9	0.01579	0.2803	19.8	119.0	138.8	.9657	1.2235
4	326.5	0.01588	0.2707	20.8	118.0	138.8	.9679	1.2224
6	337.4	0.01596	0.2614	21.8	116.9	138.7	.9701	1.2212
8	348.7	0.01605	0.2526	22.9	115.8	138.7	.9722	1.2199
10	360.2	0.01614	0.2437	24.0	114.7	138.7	.9744	1.2188
12	371.9	0.01623	0.2354	25.0	113.6	138.6	.9765	1.2175
14	383.9	0.01632	0.2274	26.1	112.5	138.6	.9787	1.2163
16	396.2	0.01642	0.2197	27.2	111.3	138.5	.9810	1.2151
18	408.9	0.01652	0.2121	28.3	110.1	138.4	.9833	1.2139
20	421.8	0.01663	0.2049	29.4	108.9	138.3	.9856	1.2127
22	434.0	0.01673	0.1979	30.5	107.7	138.2	.9879	1.2115
24	448.4	0.01684	0.1912	31.7	106.4	138.1	.9902	1.2103
26	462.2	0.01695	0.1846	32.9	105.1	138.0	.9927	1.2091
28	476.3	0.01707	0.1783	34.1	103.8	137.9	.9951	1.2079
30	490.8	0.01719	0.1722	35.4	102.4	137.8	.9976	1.2067
32	505.5	0.01731	0.1663	36.7	101.0	137.7	1.0000	1.2055
34	522.6	0.01744	0.1603	37.9	99.5	137.4	1.0023	1.2038
36	536.0	0.01759	0.1550	39.1	98.1	137.2	1.0046	1.2024
38	551.7	0.01773	0.1496	40.4	96.5	136.9	1.0071	1.2009
40	567.8	0.01787	0.1444	41.7	95.0	136.7	1.0092	1.1994
42	584.3	0.01801	0.1393	42.9	93.4	136.3	1.0115	1.1979
44	601.1	0.01817	0.1344	44.3	91.8	136.1	1.0140	1.1964
46	618.2	0.01834	0.1297	45.6	90.1	135.7	1.0166	1.1949
48	635.7	0.01851	0.1250	47.0	88.4	135.4	1.0194	1.1933
50	653.6	0.01868	0.1205	48.4	86.6	135.0	1.0218	1.1917
52	671.9	0.01887	0.1161	49.8	84.7	134.5	1.0254	1.1901
54	690.6	0.01906	0.1117	51.2	82.7	133.9	1.0272	1.1882
56	709.5	0.01927	0.1075	52.6	80.8	133.4	1.0283	1.1865
58	728.8	0.01948	0.1034	54.0	78.7	132.7	1.0312	1.1846
60	748.6	0.01970	0.0994	55.5	76.6	132.1	1.0353	1.1826
64	789.4	0.02020	0.0918	58.6	72.0	130.6	1.0410	1.1803
66	810.3	0.02048	0.0880	60.2	69.5	129.7	1.0438	1.1780
68	831.6	0.02079	0.08422	61.9	66.8	128.7	1.0468	1.1754
70	853.4	0.02112	0.08040	63.7	63.8	127.5	1.0500	1.1724
74	898.2	0.02192	0.07269	67.3	57.2	124.5	1.0568	1.1690
76	921.3	0.02242	0.06875	69.4	53.4	122.8	1.0607	1.1655
78	944.8	0.02300	0.06473	71.6	49.3	120.9	1.0649	1.1605
80	968.7	0.02370	0.06064	73.9	44.8	118.7	1.0694	1.1555
82	993.0	0.02456	0.05648	76.4	40.2	116.6	1.0740	1.1505
84	1017.7	0.02553	0.05223	79.4	34.5	113.9	1.0790	1.1455
86	1043.0	0.02686	0.04789	83.3	27.1	110.4	1.0854	1.1381
87.8	1069.4	0.03454	0.03454	97.0	0.0	97.0	1.1098	1.1098

For properties of Superheated Vapor see Fig. 25C in Cover Pocket.

TABLE 16.—SATURATED SULPHUR DIOXIDE (SO_2).

Temp.	Pressure	Volume	Density	Total heat above −40°			Entropy from −40°		
°F. t	Abs., lb./in.² p	Vapor, ft.³ lb. v_g	Liquid, lb. ft. $1/v_f$	Liquid, Btu./lb h_f	Latent, Btu./lb. h	Vapor, Btu./lb h_g	Liquid s_f	Evap.	Vapor s_g
−40	3.136	22.42	95.79	0.00	178.61	178.61	0.00000	0.42562	0.42562
−30	4.331	16.56	94.94	2.93	176.97	179.90	0.00674	0.41190	0.41864
−20	5.883	12.42	94.10	5.98	175.09	181.07	0.01366	0.39826	0.41192
−10	7.863	9.44	93.27	9.16	172.97	182.13	0.02075	0.38469	0.40544
0	10.35	7.280	92.42	12.44	170.63	183.07	0.02795	0.37122	0.39917
2	10.91	6.923	92.25	13.12	170.13	183.25	0.02941	0.36853	0.39794
4	11.50	6.584	92.08	13.78	169.63	183.41	0.03084	0.36586	0.39670
5	11.81	6.421	92.00	14.11	169.38	183.49	0.03155	0.36454	0.39609
6	12.12	6.266	91.91	14.45	169.12	183.57	0.03228	0.36319	0.39547
8	12.75	5.967	91.74	15.13	168.60	183.73	0.03373	0.36053	0.39426
10	13.42	5.682	91.58	15.80	168.07	183.87	0.03519	0.35787	0.39306
11	13.77	5.548	91.49	16.14	167.80	183.94	0.03592	0.35654	0.39246
12	14.12	5.417	91.41	16.48	167.53	184.01	0.03664	0.35521	0.39185
13	14.48	5.289	91.33	16.81	167.26	184.07	0.03737	0.35388	0.39125
14	14.84	5.164	91.24	17.15	166.97	184.14	0.03808	0.35257	0.39065
15	15.21	5.042	91.16	17.49	166.72	184.21	0.03880	0.35125	0.39005
16	15.59	4.926	91.07	17.84	166.44	184.28	0.03953	0.34993	0.38946
17	15.98	4.812	90.98	18.18	166.16	184.34	0.04026	0.34861	0.38887
18	16.37	4.701	90.89	18.52	165.88	184.40	0.04098	0.34729	0.38827
19	16.77	4.593	90.80	18.86	165.60	184.46	0.04169	0.34598	0.38767
20	17.18	4.487	90.71	19.20	165.32	184.52	0.04241	0.34466	0.38707
21	17.60	4.386	90.62	19.55	165.03	184.58	0.04313	0.34335	0.38648
22	18.03	4.287	90.53	19.90	164.74	184.64	0.04385	0.34204	0.38589
23	18.46	4.190	90.44	20.24	164.45	184.69	0.04457	0.34073	0.38530
24	18.89	4.096	90.33	20.58	164.16	184.74	0.04528	0.33943	0.38471
25	19.34	3.994	90.24	20.92	163.87	184.79	0.04600	0.33812	0.38412
26	19.80	3.915	90.15	21.26	163.58	184.84	0.04671	0.33683	0.38354
27	20.26	3.829	90.06	21.61	163.28	184.89	0.04743	0.33553	0.38296
28	20.73	3.744	89.96	21.96	162.98	184.94	0.04814	0.33422	0.38236
29	21.21	3.662	89.86	22.30	162.68	184.98	0.04886	0.33292	0.38178
30	21.70	3.581	89.76	22.64	162.38	185.02	0.04956	0.33163	0.38119
31	22.20	3.503	89.67	22.98	162.08	185.06	0.05027	0.33034	0.38061
32	22.71	3.437	89.58	23.33	161.77	185.10	0.05099	0.32904	0.38003
33	23.23	3.355	89.48	23.68	161.46	185.14	0.05171	0.32774	0.37945
34	23.75	3.283	89.39	24.03	161.15	185.18	0.05242	0.32645	0.37887
35	24.28	3.212	89.29	24.38	160.84	185.22	0.05312	0.32517	0.37829
40	27.10	2.887	88.81	26.12	159.25	185.37	0.05668	0.31873	0.37541
45	30.15	2.601	88.34	27.86	157.62	185.48	0.06020	0.31234	0.37254
50	33.45	2.348	87.87	29.61	155.95	185.56	0.06370	0.30599	0.36969
55	37.05	2.124	87.41	31.36	154.24	185.60	0.06715	0.29971	0.36686
60	40.93	1.926	86.95	33.10	152.49	185.59	0.07060	0.29345	0.36405
65	45.13	1.749	86.50	34.84	150.70	185.54	0.07401	0.28724	0.36125
70	49.62	1.590	86.02	36.58	148.88	185.46	0.07736	0.28110	0.35846
75	54.47	1.448	85.52	38.32	147.02	185.34	0.08070	0.27498	0.35568
80	59.68	1.321	85.03	40.05	145.12	185.17	0.08399	0.26897	0.35296
81	60.77	1.297	84.93	40.39	144.74	185.13	0.08462	0.26772	0.35234
82	61.88	1.274	84.84	40.73	144.36	185.09	0.08525	0.26652	0.35177
83	63.01	1.253	84.74	41.08	143.97	185.05	0.08589	0.26532	0.35121
84	64.14	1.229	84.64	41.43	143.58	185.01	0.08653	0.26412	0.35065
85	65.28	1.207	84.54	41.78	143.19	184.97	0.08718	0.26291	0.35009
86	66.45	1.185	84.44	42.12	142.80	184.92	0.08783	0.26171	0.34954
87	67.64	1.164	84.35	42.46	142.41	184.87	0.08847	0.26052	0.34899
88	68.84	1.144	84.25	42.80	142.02	184.82	0.08910	0.25933	0.34843
89	70.04	1.124	84.15	43.15	141.62	184.77	0.08974	0.25813	0.34787
90	71.25	1.104	84.05	43.50	141.22	184.72	0.09038	0.25693	0.34731
95	77.60	1.011	83.57	45.20	139.23	184.43	0.09349	0.25103	0.34452
100	84.52	0.9262	83.07	46.90	137.20	184.10	0.09657	0.24516	0.34173
110	99.76	0.7804	82.03	50.26	133.05	183.31	0.10254	0.23357	0.33611
120	120.93	0.6598	80.90	53.58	128.78	182.36	0.10829	0.22217	0.33046
140	158.61	0.4758	78.61	60.04	119.90	179.94	0.11893	0.19990	0.31883

TABLE 17.—METHYL CHLORIDE (CH_3Cl).
PROPERTIES OF SATURATED VAPOR.

Temp	Pressure		Volume		Density		Heat Content from —40°			Entropy from —40°		Temp.
°F t	Abs lb./in.² p	Gage lb./in.² p_d	Liquid ft.³/lb. v_f	Vapor ft.³/lb. v_g	Liquid lb./ft.³ $1/v_f$	Vapor lb./ft.³ $1/v_g$	Liquid Btu./lb. h_f	Latent Btu./lb. h	Vapor Btu./lb. h_g	Liquid Btu./lb.°F s_f	Vapor Btu./lb.°F s_g	°F. t
-80	1.953	25.94*	0.01493	41.08	66.98	0.02434	-13.888	198.64	184.75	-0.0351	0.4882	-80
-70	2.751	24.32*	.01508	29.84	66.31	.03351	-10.521	196.77	186.25	-0.0261	.4790	-70
-60	3.799	22.19*	.01523	22.09	65.66	.04527	- 7.039	194.78	187.74	-0.0172	.4703	-60
-50	5.155	19.43*	.01538	16.64	65.02	.06010	- 3.532	192.72	189.19	-0.0085	.4620	-50
-40	6.878	15.92*	0.01553	12.72	64.39	0.07861	0.000	190.66	190.66	0.0000	0.4544	-40
-38	7.272	15.12*	.01556	12.08	64.27	.08278	.713	190.23	190.95	.0017	.4529	-38
-36	7.684	14.28*	.01559	11.48	64.14	.08712	1.426	189.81	191.23	.0034	.4515	-36
-34	8.115	13.40*	.01562	10.91	64.02	.09166	2.138	189.38	191.51	.0051	.4500	-34
-32	8.566	12.48*	.01565	10.38	63.90	.09639	2.850	188.95	191.80	.0067	.4486	-32
-30	9.036	11.52*	0.01568	9.873	63.78	0.1013	3.562	188.52	192.08	0.0084	0.4472	-30
-28	9.526	10.53*	.01571	9.399	63.65	.1064	4.277	188.09	192.37	.0100	.4458	-28
-26	10.04	9.490*	.01574	8.953	63.53	.1117	4.993	187.65	192.65	.0117	.4445	-26
-24	10.57	8.399*	.01577	8.533	63.41	.1172	5.711	187.22	192.93	.0133	.4431	-24
-22	11.13	7.267*	.01580	8.136	63.29	.1229	6.427	186.78	193.21	.0150	.4418	-22
-20	11.71	6.090*	0.01583	7:761	63.17	0.1289	7.146	186.34	193.49	0.0166	0.4405	-20
-18	12.31	4.866*	.01566	7.408	63.86	.1350	7.863	185.90	193.76	.0183	.4393	-18
-16	12.93	3.594*	.01589	7.074	62.93	.1414	8.584	185.46	194.04	.0199	.4380	-16
-14	13.58	2.268*	.01592	6.758	62.81	.1480	9.307	185.01	194.32	.0215	.4367	-14
-12	14.26	.890*	.01596	6.459	62.70	.1548	10.03	184.56	194.59	.0232	.4355	-12
-10	14.96	.266	0.01598	6.176	62.58	0.1619	10.75	184.11	194.87	0.0247	0.4343	-10
- 8	15.69	.996	.01601	5.908	62.46	.1693	11.48	183.66	195.14	.0263	.4331	- 8
- 6	16.45	1.754	.01604	5.654	62.34	.1769	12.20	183.21	195.42	.0279	.4319	- 6
- 4	17.24	2.540	.01607	5.413	62.23	.1847	12.93	182.76	195.69	.0295	.4307	- 4
- 2	18.05	3.356	.01610	5.185	62.11	.1929	13.66	182.30	195.96	.0311	.4296	- 2
0	18.90	4.201	0.01613	4.969	62.00	0.2013	14.39	181.85	196.23	0.0327	0.4284	0
2	19.77	5.077	.01616	4.763	61.88	.2100	15.12	181.39	196.51	.0343	.4273	2
4	20.68	5.985	.01619	4.568	61.77	.2189	15.85	180.93	196.78	.0359	.4262	4
5	21.15	6.455	.01622	4.471	61.65	.2237	16.21	180.70	196.92	.0367	.4257	5
6	21.62	6.924	.01625	4.379	61.54	.2284	16.58	180.47	197.05	.0375	.4251	6
8	22.59	7.896	.01628	4.206	61.43	.2378	17.31	180.01	197.31	.0390	.4240	8
10	23.60	8.903	0.01631	4.038	61.31	0.2477	18.04	179.53	197.58	0.0406	0.4229	10
12	24.64	9.943	.01634	3.878	61.20	.2579	18.77	179.06	197.83	.0422	.4218	12
14	25.72	11.02	.01637	3.726	61.09	.2684	19.51	178.58	198.09	.0437	.4208	14
16	26.83	12.13	.01640	3.581	60.98	.2792	20.25	178.10	198.34	.0453	.4198	16
18	27.97	13.28	.01644	3.443	60.83	.2904	20.98	177.61	198.59	.0468	.4187	18
20	29.16	14.46	0.01647	3.312	60.72	0.3019	21.73	177.11	198.84	0.0484	0.4177	20
22	30.38	15.69	.01650	3.186	60.61	.3138	22.47	176.61	199.08	.0499	.4166	22
24	31.64	16.95	.01654	3.067	60.46	.3261	23.21	176.11	199.32	.0514	.4156	24
26	32.95	18.25	.01658	2.952	60.31	.3388	23.95	175.61	199.56	.0530	.4146	26
28	34.29	19.60	.01662	2.843	60.17	.3517	24.70	175.10	199.79	.0545	.4136	28
30	35.68	20.98	0.01665	2.739	60.06	0.3650	25.44	174.59	200.03	0.0560	0.4126	30
32	37.11	22.41	.01669	2.640	59.92	.3787	26.18	174.08	200.26	.0575	.4117	32
34	38.58	23.88	.01673	2.546	59.77	.3928	26.93	173.56	200.49	.0590	.4107	34
36	40.09	25.39	.01677	2.455	59.63	.4073	27.67	173.05	200.72	.0605	.4098	36
38	41.65	26.95	.01681	2.369	59.49	.4222	28.42	172.53	200.95	.0621	.4088	38
40	43.25	28.56	0.01684	2.286	59.38	0.4375	29.17	172.00	201.17	0.0636	0.4079	40
42	44.91	30.21	.01688	2.206	59.24	.4532	29.92	171.48	201.40	.0651	.4070	42
44	46.61	31.91	.01692	2.130	59.10	.4694	30.67	170.95	201.62	.0665	.4061	44
46	48.35	33.66	.01696	2.057	58.96	.4861	31.42	170.42	201.84	.0680	.4052	46
48	50.15	35.45	.01700	1.987	58.82	.5033	32.17	169.89	202.06	.0695	.4043	48

* Inches of mercury below one atmosphere.

TABLE 17.—METHYL CHLORIDE (CH_3Cl).
PROPERTIES OF SATURATED VAPOR.—(Concluded.)

Temp.	Pressure		Volume		Density		Heat Content from $-40°$			Entropy from $-40°$		Temp.
°F. t	Abs. lb./in.² p	Gage lb./in.² p_d	Liquid ft.³/lb. v_f	Vapor ft.³/lb. v_g	Liquid lb./ft.³ $1/v_f$	Vapor lb./ft.³ $1/v_g$	Liquid Btu./lb. h_f	Latent Btu./lb. h	Vapor Btu./lb. h_g	Liquid Btu./lb. °F. s_f	Vapor Btu./lb. °F. s_g	°F. t
50	51.99	37.29	0.01704	1.920	58.69	0.5208	32.93	169.35	202.28	0.0710	0.4034	50
52	53.88	39.18	.01708	1.856	58.55	.5388	33.68	168.81	202.49	.0725	.4025	52
54	55.83	41.13	.01712	1.794	58.41	.5573	34.44	168.27	202.71	.0740	.4017	54
56	57.83	43.13	.01716	1.735	58.28	.5763	35.19	167.72	202.91	.0754	.4008	56
58	59.88	45.19	.01720	1.679	58.14	.5958	35.95	167.18	203.13	.0769	.3999	58
60	62.00	47.30	0.01724	1.624	58.00	0.6158	36.71	166.62	203.33	0.0784	0.3991	60
62	64.17	49.47	.01728	1.572	57.87	.6362	37.47	166.07	203.54	.0798	.3983	62
64	66.39	51.70	.01732	1.522	57.74	.6572	38.23	165.51	203.74	.0813	.3974	64
66	68.67	53.98	.01736	1.473	57.60	.6788	39.00	164.95	203.95	.0827	.3966	66
68	71.01	56.32	.01740	1.427	57.47	.7008	39.76	164.39	204.15	.0842	.3958	68
70	73.41	58.71	0.01744	1.382	57.34	0.7234	40.52	163.82	204.34	0.0856	0.3950	70
72	75.86	61.17	.01748	1.339	57.21	.7467	41.29	163.24	204.53	.0870	.3941	72
74	78.37	63.68	.01752	1.298	57.08	.7704	42.06	162.66	204.72	.0885	.3933	74
76	80.94	66.25	.01756	1.258	56.95	.7948	42.82	162.08	204.90	.0899	.3925	76
78	83.57	68.87	.01760	1.220	56.82	.8196	43.59	161.50	205.09	.0913	.3918	78
80	86.26	71.56	0.01764	1.183	56.69	0.8451	44.36	160.91	205.27	0.0928	0.3910	80
82	89.01	74.31	.01768	1.148	56.56	.8710	45.13	160.32	205.45	.0942	.3902	82
84	91.82	77.13	.01773	1.114	56.40	.8979	45.90	159.72	205.62	.0956	.3894	84
86	94.70	80.00	.01778	1.081	56.24	.9253	46.67	159.13	205.80	.0970	.3887	86
88	97.64	82.94	.01782	1.049	56.12	.9531	47.44	158.52	205.96	.0984	.3879	88
90	100.6	85.95	0.01786	1.018	55.99	0.9819	48.21	157.92	206.13	0.0998	0.3872	90
92	103.7	89.02	.01791	.9889	55.83	1.011	48.99	157.31	206.30	.1012	.3865	92
94	106.9	92.16	.01796	.9603	55.68	1.041	49.77	156.69	206.46	.1026	.3857	94
96	110.1	95.37	.01800	.9333	55.56	1.072	50.54	156.08	206.62	.1041	.3850	96
98	113.4	98.65	.01804	.9069	55.43	1.103	51.32	155.46	206.78	.1055	.3843	98
100	116.7	102.0	0.01808	.8814	55.31	1.135	52.09	154.85	206.94	0.1069	0.3836	100
102	120.1	105.4	.01813	.8568	55.15	1.167	52.87	154.22	207.09	.1082	.3828	102
104	123.6	108.9	.01818	.8331	55.01	1.200	53.65	153.60	207.25	.1096	.3822	104
106	127.2	112.5	.01823	.8105	54.85	1.234	54.43	152.97	207.40	.1110	.3815	106
108	130.8	116.1	.01828	.7884	54.70	1.268	55.22	152.33	207.55	.1124	.3808	108
110	134.5	119.8	0.01833	.7672	54.55	1.303	56.00	151.70	207.70	0.1138	0.3801	110
112	138.3	123.6	.01838	.7466	54.41	1.339	56.78	151.06	207.84	.1151	.3794	112
114	142.2	127.5	.01843	.7268	54.26	1.376	57.57	150.41	207.98	.1165	.3787	114
116	146.1	131.4	.01848	.7075	54.11	1.414	58.36	149.77	208.13	.1179	.3781	116
118	150.1	135.4	.01853	.6889	53.97	1.452	59.15	149.11	208.26	.1193	.3774	118
120	154.2	139.5	0.01859	.6710	53.79	1.490	59.93	148.46	208.39	0.1206	0.3768	120
122	158.4	143.7	.01865	.6534	53.62	1.530	60.73	147.80	208.53	.1220	.3762	122
124	162.6	147.9	.01870	.6367	53.48	1.571	61.51	147.14	208.65	.1234	.3755	124
126	167.0	152.3	.01876	.6201	53.33	1.613	62.31	146.47	208.78	.1247	.3749	126
128	171.4	156.7	.01881	.6043	53.16	1.655	63.10	145.80	208.90	.1261	.3742	128
130	175.9	161.1	0.01887	.5889	52.99	1.698	63.89	145.13	209.02	0.1274	0.3736	130
132	180.4	165.7	.01893	.5741	52.83	1.742	64.69	144.45	209.14	.1288	.3730	132
134	185.1	170.4	.01898	.5596	52.69	1.787	65.48	143.77	209.25	.1301	.3723	134
136	189.8	175.1	.01904	.5455	52.52	1.833	66.28	143.09	209.37	.1314	.3717	136
138	194.7	180.0	.01909	.5320	52.38	1.880	67.08	142.40	209.48	.1328	.3711	138
140	199.6	184.9	0.01915	.5189	52.22	1.927	67.87	141.71	209.58	0.1341	0.3705	140
150	225.4	210.7	0.01945	.4586	51.41	2.181	71.87	138.23	210.10	0.1407	0.3674	150
160	253.5	238.8	0.01978	.4070	50.56	2.457	75.90	134.66	210.56	0.1473	0.3646	160
170	283.9	269.2	0.02015	.3613	49.63	2.768	79.97	130.96	210.93	0.1538	0.3618	170

TABLE 18.—PROPERTIES OF SOLUTIONS OF AMMONIA AND WATER.

(Figures are boiling points due to corresponding pressures and concentrations.)
From "Mechanical Refrigeration," by H. J. Macintire (John Wiley & Co.)

Per cent, NH_3.	Absolute pressure.																
	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
2	169.0	173.0	178.5	181.8	185.6	189.5	193.2	197.0	201.5	203.5	206.5	209.5	212.3	215.0	217.5	220.1	222.4
4	161.0	165.4	169.5	173.3	177.0	180.9	184.7	188.4	191.5	194.6	197.5	200.9	203.0	205.6	207.9	210.5	212.8
6	153.0	157.0	161.0	165.0	168.5	172.5	176.0	179.1	182.4	185.7	188.8	191.4	194.0	196.5	199.0	201.4	203.7
8	143.5	148.3	152.7	156.4	160.5	164.0	167.9	170.8	174.0	177.9	180.1	182.7	185.3	188.0	190.3	192.5	195.0
10	137.4	141.0	144.5	148.4	151.8	155.0	158.8	162.0	165.0	168.0	171.1	173.9	176.5	179.1	181.4	183.7	186.0
12	129.0	132.5	136.7	140.6	144.3	147.8	151.3	154.6	157.6	160.4	163.1	165.6	168.0	170.6	172.9	175.0	177.2
14	120.5	124.6	129.0	133.0	136.7	140.0	143.6	147.0	150.0	152.7	155.5	158.0	160.6	162.9	165.1	167.4	169.7
16	113.0	117.0	121.4	125.4	129.5	132.6	136.0	139.0	142.0	144.6	147.4	149.8	152.1	154.6	156.9	159.0	161.3
18	107.5	110.0	113.5	118.0	121.5	125.0	128.1	131.3	134.4	137.0	139.7	142.3	144.7	147.0	149.2	151.4	153.5
20	99.0	103.0	106.8	110.5	114.0	117.6	120.6	129.0	126.7	129.5	132.0	134.5	137.0	139.3	141.6	143.7	145.7
22	92.0	95.5	99.2	103.0	106.5	110.0	113.3	116.2	119.1	122.0	124.6	127.0	129.5	131.6	133.9	136.0	138.0
24	86.0	89.7	93.5	97.0	100.0	103.1	106.0	109.0	111.8	114.5	117.0	119.6	122.4	124.8	127.0	129.4	131.5
26	79.2	82.6	86.0	90.0	93.0	96.0	99.5	102.4	105.3	108.0	110.6	113.2	115.6	118.1	120.1	122.1	124.1
28	73.4	77.0	81.0	84.4	87.8	90.7	83.5	96.0	98.8	101.3	103.6	106.2	108.7	111.0	113.3	115.5	117.3
30	66.5	70.6	74.5	77.5	80.9	83.8	86.7	89.6	92.1	94.7	97.0	99.3	101.8	104.0	106.3	108.5	110.5
32	61.0	64.5	68.0	71.0	74.0	77.0	79.8	82.3	85.0	87.5	90.0	92.3	94.5	96.9	99.0	101.0	103.0
34	55.0	59.0	62.4	65.6	68.8	71.6	74.3	76.8	79.4	81.7	84.0	86.5	88.5	90.6	92.8	94.8	96.8
36	49.0	52.7	56.5	60.0	63.0	66.0	69.0	71.7	73.8	76.1	78.5	80.5	82.6	84.7	86.8	88.9	90.8
38	43.0	47.0	50.5	54.0	57.0	60.0	63.0	65.5	68.0	70.2	72.5	74.5	76.6	78.7	80.7	82.7	84.6
40	37.0	41.0	44.5	48.0	51.0	53.5	56.2	59.5	61.0	63.5	66.0	68.2	70.4	72.6	74.8	76.9	79.0
42	31.6	35.7	39.4	42.9	46.0	48.6	51.4	53.9	56.3	58.7	61.0	63.2	65.2	67.3	69.1	71.0	73.9
44	27.5	30.2	34.0	37.0	40.5	43.5	45.9	48.5	50.8	53.0	55.3	57.4	59.8	61.6	63.5	65.4	67.1
46	21.2	24.9	28.5	31.7	34.9	37.6	40.4	43.0	45.4	47.6	49.7	51.7	53.8	56.0	57.8	59.8	61.5
48	16.0	19.8	23.0	26.0	29.0	32.0	34.9	37.4	40.0	42.2	44.4	46.5	48.5	50.5	52.4	54.2	56.0
50	11.0	14.6	18.0	21.0	24.0	26.9	29.7	32.0	34.4	36.6	38.9	40.9	42.8	44.7	46.6	48.5	50.3

TABLE 18.—PROPERTIES OF SOLUTIONS OF AMMONIA AND WATER.—(Continued.)

Per cent. NH_3	Absolute pressure.																
	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43
2	224.6	226.9	229.0	231.1	233.1	235.1	237.0	238.9	240.7	242.4	244.2	246.0	247.6	249.4	251.0	252.5	254.2
4	215.3	217.7	219.9	221.9	224.0	226.0	228.0	229.8	231.6	233.4	235.3	237.0	238.7	240.3	242.0	243.4	245.0
6	205.0	208.1	210.6	212.4	214.5	216.4	218.3	220.1	222.0	223.5	225.0	226.5	228.0	229.5	231.0	232.0	233.5
8	497.3	199.5	201.7	203.7	205.6	207.5	209.3	211.3	213.0	214.9	216.6	218.1	219.9	221.4	223.0	224.5	226.0
10	188.4	190.5	192.7	194.6	196.6	198.7	200.5	202.3	204.1	205.9	207.3	209.0	210.6	212.3	213.8	215.2	216.8
12	179.4	181.5	183.4	185.5	187.4	189.4	191.2	193.0	195.0	196.7	198.4	200.0	201.7	203.3	204.9	206.3	207.9
14	171.8	173.6	175.8	177.6	179.5	181.4	183.2	185.0	186.8	188.5	190.2	192.0	193.6	195.2	196.9	198.2	199.7
16	163.4	165.4	167.4	169.4	171.3	173.2	175.1	177.0	178.9	180.6	182.3	183.7	185.4	187.1	188.6	190.0	191.4
18	155.5	157.4	159.9	161.2	163.0	165.0	166.8	168.4	170.0	171.8	173.4	175.1	176.7	178.3	179.9	181.1	182.7
20	147.8	149.8	151.8	153.8	155.7	157.6	159.3	161.0	162.5	164.3	165.9	167.3	168.7	170.2	171.5	172.8	174.2
22	140.0	142.0	144.0	146.0	147.9	149.7	151.5	153.2	155.0	156.5	158.0	159.6	161.0	162.7	164.0	165.5	167.0
24	133.5	135.4	137.3	139.2	141.0	142.8	144.4	146.0	147.6	149.2	150.8	152.2	153.7	155.2	156.6	158.0	159.4
26	126.1	128.1	130.0	131.9	133.7	135.5	137.2	138.9	140.4	142.0	143.6	145.2	146.6	148.2	149.5	151.0	152.2
28	119.4	121.3	123.0	125.0	126.6	128.3	129.8	131.5	133.1	134.6	136.0	137.5	139.0	140.6	142.0	143.4	144.8
30	112.5	114.4	116.0	117.8	119.6	121.2	122.8	124.5	126.0	127.4	129.0	130.5	132.0	133.5	134.7	136.0	137.4
32	105.0	107.0	109.0	111.0	113.0	114.9	116.7	118.0	119.8	121.3	122.8	124.3	125.5	127.1	128.3	129.8	131.0
34	98.8	100.8	102.7	104.5	106.3	108.0	109.6	111.3	112.8	114.5	115.8	117.3	118.5	120.0	121.3	122.5	123.8
36	92.7	94.4	96.2	98.1	100.0	101.7	103.2	104.9	106.5	108.0	109.5	111.0	112.3	113.5	115.0	116.2	117.5
38	86.4	88.2	90.0	91.8	93.5	95.1	96.6	98.2	99.5	101.0	102.5	104.0	105.4	106.8	108.0	109.2	110.5
40	80.2	82.2	84.1	86.0	87.7	89.2	90.8	92.4	93.7	95.3	96.7	98.0	99.4	101.0	102.0	103.5	104.7
42	74.5	76.1	77.8	79.4	81.0	82.5	84.0	85.5	86.9	88.2	89.7	91.2	92.6	94.0	95.0	96.5	98.0
44	68.8	70.4	72.0	73.6	75.1	76.7	78.2	79.6	81.1	82.4	83.7	85.0	86.5	87.8	89.0	90.4	91.7
46	63.3	65.0	66.5	68.1	69.6	71.2	72.7	74.0	75.4	76.7	78.2	79.6	80.9	82.0	83.0	84.5	85.7
48	58.0	59.8	61.3	62.7	64.3	65.8	67.3	68.7	70.3	71.5	72.8	74.2	75.5	76.7	78.0	79.0	80.4
50	52.0	53.5	55.1	56.8	58.3	59.9	61.3	62.5	64.1	65.5	66.9	68.3	69.7	70.9	72.2	73.4	74.7

TABLE 18.—PROPERTIES OF SOLUTIONS OF AMMONIA AND WATER.—(Continued.)

Per cent. NH_3.	Absolute pressure.																
	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60
2	255.6	257.1	258.6	260.0	261.5	262.9	264.3	265.8	267.1	268.4	269.8	271.2	272.4	273.8	274.9	276.0	277.9
4	246.4	247.9	249.5	250.8	252.2	253.6	255.0	256.4	257.9	259.2	260.4	261.7	262.9	264.0	265.3	266.6	267.7
6	235.0	236.5	238.0	239.6	241.0	242.5	244.0	246.4	248.0	249.3	250.6	251.8	253.0	254.4	255.5	256.7	258.0
8	227.4	228.7	230.3	231.7	233.1	234.5	236.0	237.3	238.6	239.8	241.0	242.3	243.5	244.7	245.8	247.0	248.2
10	218.3	219.8	221.2	222.8	224.3	225.7	227.0	228.3	229.7	230.8	232.3	233.6	234.8	236.0	237.3	238.4	239.6
12	209.3	210.7	211.9	213.3	214.6	215.9	217.2	218.4	219.8	221.0	222.4	223.5	224.6	225.7	226.8	228.0	229.2
14	201.1	202.5	204.0	205.3	206.5	207.8	209.1	210.2	211.7	212.4	214.0	215.2	216.3	217.4	218.5	219.7	220.8
16	192.9	194.1	195.5	196.8	198.0	199.4	200.5	201.8	203.0	204.1	205.3	206.5	207.7	208.8	209.8	210.9	212.0
18	184.0	185.5	186.8	188.1	189.4	190.7	192.0	193.1	194.4	195.5	196.7	197.8	198.9	200.0	201.0	202.0	203.1
20	175.5	177.0	178.5	179.9	181.0	182.2	183.6	184.8	186.0	187.2	188.4	189.7	190.8	192.0	193.2	194.2	195.4
22	168.3	169.5	170.9	172.1	173.4	174.6	175.9	177.0	178.2	179.5	180.6	181.7	182.8	183.9	184.9	185.9	187.0
24	160.8	162.0	163.5	164.9	166.1	167.3	168.7	170.0	171.2	172.4	173.6	174.8	175.8	177.0	178.0	179.0	180.0
26	153.5	155.0	156.2	157.5	158.6	159.9	161.0	162.5	163.5	164.5	165.6	166.7	167.8	169.0	170.0	171.0	172.1
28	146.0	147.4	148.9	150.1	151.5	152.7	154.0	155.2	156.5	157.6	158.8	160.0	161.0	162.0	163.0	164.1	165.2
30	138.8	140.1	141.3	142.6	143.8	145.1	146.1	147.3	148.4	149.7	150.8	151.8	152.9	153.9	154.9	155.9	157.0
32	132.1	133.5	134.7	135.8	137.0	138.1	139.4	140.4	141.4	142.5	143.5	144.6	145.6	146.6	147.6	148.7	149.7
34	125.0	126.2	127.5	128.6	129.9	131.0	132.0	133.0	134.1	135.2	136.3	137.3	138.3	139.5	140.5	141.4	142.4
36	118.7	120.0	121.1	122.4	123.5	124.7	125.8	126.9	127.9	129.0	130.0	131.0	132.0	133.0	134.0	135.0	136.0
38	111.9	113.2	114.5	115.7	117.0	118.1	119.4	120.5	121.7	122.8	123.9	125.0	126.0	127.0	128.0	129.0	130.0
40	105.9	107.0	108.1	109.5	110.6	111.8	112.9	114.0	115.2	116.3	117.3	118.3	119.4	120.5	121.5	122.4	123.4
42	99.2	100.5	101.7	193.0	104.0	105.1	106.3	107.4	108.5	109.6	110.6	111.6	112.5	115.5	114.5	115.5	116.4
44	93.0	94.3	95.5	96.7	97.9	99.0	100.1	101.2	102.3	103.4	104.4	105.5	106.5	107.4	108.3	109.3	110.3
46	86.9	88.1	89.4	90.5	91.7	92.8	94.0	95.0	96.1	97.2	98.3	99.4	100.3	101.3	102.3	103.3	104.2
48	81.5	82.6	83.8	84.9	86.0	87.1	88.3	89.4	90.4	91.5	92.5	93.5	94.5	95.5	96.5	97.5	98.5
50	75.9	77.0	78.1	79.3	80.4	81.4	82.6	83.7	84.8	85.9	86.9	87.9	88.8	89.8	90.8	91.7	92.6

TABLE 18.—PROPERTIES OF SOLUTIONS OF AMMONIA AND WATER.—(Continued.)

Per cent, NH_3	Absolute pressure.																
	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77
2	278.6	279.7	280.8	282.0	283.2	284.3	285.4	286.5	287.7	288.8	290.0	291.0	292.0	293.0	294.0	295.0	296.0
4	268.9	270.2	271.2	272.3	273.5	274.6	175.7	276.9	277.9	278.9	280.0	281.0	282.0	283.0	284.0	285.0	285.9
6	259.1	260.3	261.4	262.5	263.6	264.8	265.9	267.0	268.0	269.0	270.0	271.0	272.0	273.0	274.0	274.9	275.9
8	249.4	250.6	251.6	252.7	253.8	255.0	256.0	257.1	258.1	259.0	260.0	261.0	262.0	263.0	264.0	264.9	265.8
10	240.9	242.0	243.0	244.1	245.2	246.3	247.4	248.4	249.5	250.5	251.3	252.3	253.2	254.1	255.0	255.9	256.8
12	230.2	231.5	232.5	233.5	234.5	235.7	236.8	237.9	238.9	240.0	240.9	241.8	242.7	243.7	244.6	245.5	246.5
14	221.8	222.9	223.9	225.0	226.0	227.0	228.0	229.0	230.1	231.2	232.0	233.0	234.0	234.9	235.8	236.8	237.7
16	213.0	214.0	215.1	216.1	217.2	218.2	219.3	220.3	221.3	222.3	223.1	224.1	225.1	226.0	227.0	227.9	228.8
18	204.2	205.2	206.3	207.3	208.4	209.5	210.5	211.5	212.5	213.5	214.3	215.3	216.3	217.2	218.1	219.0	220.0
20	196.4	197.4	198.4	199.4	200.3	201.3	202.2	203.1	204.1	205.0	205.7	206.7	207.6	208.3	209.2	210.2	211.1
22	188.0	189.0	190.0	191.1	192.1	193.1	194.0	195.0	196.0	196.9	197.8	198.7	199.7	200.5	201.4	202.3	203.1
24	181.2	182.2	183.1	184.1	185.1	186.0	187.0	188.0	189.0	190.0	190.7	191.6	192.5	193.4	194.2	195.1	196.0
26	173.1	174.2	175.2	176.2	177.2	178.2	179.2	180.1	181.0	182.0	183.0	184.0	184.9	185.7	186.5	187.3	188.2
28	166.3	167.3	168.2	169.1	170.0	171.0	172.0	172.9	173.8	174.7	175.4	176.3	177.3	178.0	178.8	179.7	180.4
30	158.0	159.0	159.9	160.9	161.9	162.8	163.7	164.6	165.5	166.4	167.2	168.1	169.0	169.9	170.8	171.7	172.4
32	150.7	151.5	152.5	153.4	154.2	155.2	156.1	157.0	157.9	158.8	159.7	160.5	161.4	162.3	163.1	164.0	164.9
34	143.4	144.4	145.3	146.2	147.2	148.1	149.0	150.0	151.0	151.9	152.7	153.6	154.5	155.3	156.0	157.0	157.8
36	137.0	137.9	138.9	139.8	140.8	141.7	142.6	143.5	144.4	145.3	146.0	146.9	147.8	148.6	149.5	150.2	151.0
38	131.0	131.9	132.8	133.7	134.6	135.5	136.4	137.3	138.1	138.7	139.6	140.5	141.3	142.0	142.8	143.6	144.3
40	124.3	125.2	126.1	127.0	128.0	128.9	129.8	130.5	131.4	132.3	133.0	133.9	134.5	135.4	136.2	137.0	137.8
42	117.4	118.3	119.1	120.0	120.9	121.7	122.5	123.3	124.1	124.0	125.6	126.4	127.2	128.0	128.8	129.5	130.3
44	111.2	112.0	112.9	113.8	114.7	115.5	116.3	117.1	118.0	118.8	119.5	120.3	121.0	121.9	122.6	123.3	124.0
46	105.1	106.0	106.8	107.7	108.5	109.4	110.3	111.0	111.8	112.6	113.3	114.1	115.0	115.6	116.5	117.2	117.9
48	99.4	100.2	101.1	101.9	102.8	103.6	104.4	105.3	106.0	106.7	107.5	108.2	109.0	109.8	110.4	111.2	111.8
50	93.5	94.4	95.3	96.1	97.0	97.8	98.5	99.3	100.0	100.8	101.6	102.5	103.2	104.0	104.7	105.4	106.3

TABLE 18.—PROPERTIES OF SOLUTIONS OF AMMONIA AND WATER.—(Continued.)

Per cent. NH_3.	Absolute pressure.																
	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93	94
2	297.0	298.0	299.0	299.9	300.8	301.7	302.6	303.5	304.4	305.3	306.1	306.8	307.8	308.7	309.5	310.4	311.1
4	286.8	287.7	288.6	289.5	290.4	291.3	292.1	293.0	293.9	294.7	295.6	296.5	297.3	298.1	299.0	299.8	300.7
6	276.8	277.7	278.5	279.5	280.4	281.3	282.2	283.0	283.9	284.8	285.6	286.5	287.4	288.1	289.0	289.8	290.5
8	266.7	267.8	268.5	269.9	270.3	271.2	272.0	272.9	273.8	274.6	275.5	276.4	277.3	278.0	278.9	279.7	280.5
10	257.8	258.7	259.5	260.3	261.2	262.0	262.9	263.8	264.6	265.3	266.2	267.0	267.9	268.6	269.5	270.3	271.1
12	247.5	248.4	249.4	250.3	251.2	252.0	252.9	253.8	254.7	255.6	256.5	257.3	258.1	258.9	259.7	260.5	261.2
14	238.6	239.5	240.5	241.4	242.2	243.0	243.9	244.8	245.7	246.4	247.2	248.1	248.9	249.6	250.4	251.1	251.9
16	229.7	230.6	231.6	232.3	233.2	234.0	235.0	235.9	236.7	237.4	238.2	239.0	239.9	240.7	241.4	242.2	242.9
18	220.9	221.8	222.6	223.5	224.3	225.1	226.0	226.8	227.7	228.5	229.3	230.1	230.9	231.6	232.4	233.2	233.9
20	212.0	212.9	213.8	214.6	215.3	216.1	216.9	217.8	218.6	219.3	220.1	221.0	221.8	222.6	223.3	224.0	224.7
22	204.0	204.8	205.6	206.4	207.1	208.0	208.8	209.6	210.4	211.1	211.9	212.7	213.5	214.3	215.0	215.7	216.4
24	196.9	197.7	198.4	199.2	200.0	200.8	201.5	202.2	203.0	203.7	204.5	205.2	206.0	206.6	207.3	208.1	208.9
26	189.0	189.9	190.7	191.5	192.3	193.0	193.7	194.5	195.3	196.0	196.8	197.5	198.2	198.9	199.7	200.4	201.0
28	181.2	182.0	182.8	183.6	184.3	185.1	185.9	186.7	187.3	188.0	188.8	189.5	190.3	190.9	191.6	192.3	193.0
30	173.1	174.0	174.9	175.8	176.6	177.2	178.1	178.9	179.8	180.4	181.2	182.0	182.7	183.4	184.0	184.8	185.4
32	165.8	166.5	167.3	168.1	169.0	169.8	170.6	171.3	172.0	172.7	173.5	174.2	175.0	175.8	176.5	177.2	177.9
34	158.6	159.4	160.2	161.0	161.7	162.5	163.3	164.0	164.7	165.5	166.3	167.0	167.8	168.5	169.3	169.9	170.7
36	151.8	152.5	153.3	154.0	154.7	155.5	156.3	157.0	157.7	158.5	159.2	160.0	160.7	161.4	162.0	163.5	163.5
38	145.1	145.9	146.7	147.3	148.1	148.9	149.6	150.4	151.1	151.8	152.4	153.1	153.8	154.6	155.2	155.8	256.4
40	138.5	139.2	140.0	140.7	141.4	142.2	142.7	143.5	144.2	144.9	145.6	146.2	147.0	147.7	148.3	149.0	149.7
42	131.0	131.8	132.5	133.3	134.0	134.7	135.5	136.1	136.9	137.5	138.3	139.0	139.8	140.4	141.0	141.8	142.5
44	124.8	125.5	126.3	127.0	127.8	128.4	129.2	129.9	130.6	131.3	131.9	132.6	133.4	134.0	134.7	135.2	136.0
46	118.7	119.3	120.1	120.7	121.5	122.2	122.9	123.5	124.2	125.0	125.7	126.3	126.9	127.6	128.3	128.9	129.6
48	112.5	113.2	114.0	114.6	115.3	116.0	116.7	117.3	118.0	118.7	119.3	120.0	120.6	121.3	121.9	122.6	123.2
50	106.9	107.5	108.3	109.0	109.8	110.3	111.0	111.7	112.2	113.0	113.7	114.3	114.9	115.5	116.2	116.9	117.3

TABLE 18.—PROPERTIES OF SOLUTIONS OF AMMONIA AND WATER.—(Continued.)

Per cent. NH_3.	Absolute pressure.																
	95	96	97	98	99	100	101	102	103	104	105	106	107	108	109	110	111
2	312.0	312.8	313.7	314.5	315.4	316.2	316.8	317.7	318.5	319.3	320.0	320.8	321.5	322.3	323.0	323.7	324.5
4	301.5	302.3	303.1	304.0	304.7	305.6	306.3	307.1	308.1	308.7	309.5	310.3	311.0	311.9	312.6	313.4	314.1
6	291.3	292.1	292.8	293.6	294.4	295.3	295.9	296.6	297.4	298.1	298.9	299.6	200.4	301.2	301.9	302.6	303.4
8	281.3	282.0	282.8	283.5	284.4	285.1	285.8	286.7	287.4	288.2	289.0	289.7	290.4	291.2	291.9	292.5	293.4
10	271.9	272.8	273.5	274.3	275.0	275.8	276.4	277.1	278.1	278.7	279.4	280.1	280.9	281.4	282.3	282.9	283.5
12	262.0	262.7	263.4	264.1	264.9	265.6	266.4	267.0	267.8	268.5	269.2	269.9	270.5	271.3	272.0	272.7	273.4
14	252.6	253.4	254.0	254.7	255.5	256.1	256.0	256.6	257.3	259.0	259.8	260.4	261.1	261.8	262.5	263.0	263.9
16	243.7	244.7	245.0	245.7	246.5	247.3	248.0	248.7	249.3	250.0	250.7	251.4	252.0	252.7	253.4	254.0	254.7
18	234.6	235.3	236.0	236.6	237.2	238.0	238.5	239.4	240.0	240.7	241.5	242.0	242.7	243.5	244.0	244.7	245.3
20	225.5	226.3	227.0	227.8	228.5	229.1	229.9	230.5	231.2	231.9	232.6	233.2	233.9	234.5	235.2	235.9	236.5
22	217.1	217.9	218.5	219.2	220.0	220.6	221.1	222.0	222.7	223.2	224.0	224.5	225.3	225.9	226.5	227.1	227.8
24	209.7	210.4	211.0	211.7	212.4	213.0	213.7	214.3	215.0	215.6	216.1	216.8	217.4	218.0	218.7	219.2	220.0
26	201.8	202.4	203.1	203.8	204.5	205.1	205.6	206.4	207.1	207.8	208.4	209.0	209.7	210.3	211.0	211.5	212.0
28	193.6	194.4	195.0	195.7	196.4	197.1	197.6	198.4	199.0	199.6	200.3	200.9	201.4	202.0	202.5	203.1	203.8
30	186.1	186.9	187.4	188.0	188.7	189.4	189.9	190.6	191.3	191.9	193.6	193.2	193.9	194.4	105.0	195.6	196.1
32	178.6	179.3	180.0	180.6	181.3	182.0	182.5	183.1	183.9	184.5	185.1	185.8	186.3	186.9	187.6	188.2	188.8
34	171.3	172.0	172.6	173.3	174.0	174.6	175.0	175.9	176.5	177.2	177.8	178.4	179.0	179.5	180.4	180.9	181.5
36	164.1	164.8	165.5	166.0	166.7	167.3	168.0	168.6	169.3	169.8	170.5	171.0	171.7	172.3	172.9	173.5	174.0
38	157.0	157.7	158.4	159.0	159.7	160.3	160.9	161.4	162.1	162.7	163.3	164.0	164.5	165.2	165.8	166.4	167.0
40	150.3	150.8	151.5	152.1	152.8	153.4	154.0	154.6	155.2	155.8	156.4	157.0	157.6	158.1	158.8	159.4	160.0
42	143.1	143.8	144.5	145.2	145.8	146.4	147.0	147.8	148.3	149.0	149.7	150.3	150.9	151.5	152.0	152.7	153.3
44	136.6	137.2	137.9	138.5	139.2	139.9	140.5	141.1	141.8	142.4	143.0	143.7	144.3	144.8	145.4	146.0	146.6
46	130.3	130.8	131.5	132.1	132.9	133.4	134.0	134.7	135.2	133.9	136.5	137.0	137.7	138.3	138.9	139.5	140.0
48	123.8	124.5	125.1	125.7	126.3	127.0	127.5	128.1	128.7	129.3	130.0	130.6	131.1	131.6	132.3	132.9	133.9
50	118.0	118.6	119.2	119.9	120.4	121.0	121.5	122.1	122.8	123.4	123.8	124.3	125.0	125.5	126.3	126.9	127.5

TABLE 18.—PROPERTIES OF SOLUTIONS OF AMMONIA AND WATER.—(Continued.)

Per cent, NH_3.	Absolute pressure.																
	112	113	114	115	116	117	118	119	120	121	122	123	124	125	126	127	128
2	325.2	325.9	326.5	327.3	328.0	328.6	329.4	330.0	330.7	331.3	332.0	332.7	333.4	334.0	334.7	335.4	336.0
4	314.9	315.6	316.3	317.0	317.7	318.4	319.0	319.6	320.3	321.0	321.7	322.3	323.0	323.6	324.2	324.8	325.5
6	304.1	304.8	305.5	306.2	306.9	307.5	308.2	308.9	309.6	310.2	310.9	311.5	312.2	312.8	313.5	314.0	314.7
8	294.2	294.9	295.5	296.2	296.9	297.5	298.2	298.9	299.6	300.3	300.9	301.5	302.1	302.7	303.4	304.0	304.6
10	284.3	284.9	285.5	286.2	286.9	287.5	288.2	288.8	289.4	290.3	290.8	291.4	291.9	292.7	293.2	293.8	294.4
12	274.1	274.7	275.4	276.0	276.8	277.4	278.0	278.7	279.3	280.0	280.6	281.3	281.8	282.5	283.1	283.7	284.3
14	264.5	265.1	265.9	266.5	267.1	267.8	268.4	269.1	269.7	270.4	271.0	271.7	272.3	272.9	273.5	274.1	274.7
16	255.3	256.0	256.6	257.3	257.9	258.5	259.1	259.8	260.4	261.0	261.6	262.3	262.9	263.5	264.0	264.7	265.3
18	246.0	246.7	247.3	248.0	248.5	249.1	249.8	250.5	251.0	251.7	252.2	252.8	253.4	254.0	254.6	255.3	255.8
20	237.1	237.8	238.3	239.0	239.6	240.3	240.9	241.5	242.0	242.7	243.3	243.7	244.4	245.0	245.0	246.8	256.8
22	228.4	229.0	229.7	230.3	230.9	231.6	232.2	232.8	233.4	234.0	234.5	235.1	235.9	236.3	237.0	237.5	238.1
24	220.7	221.2	221.7	222.4	223.0	223.7	224.2	224.8	225.5	226.0	226.6	227.1	227.8	228.4	228.9	229.4	230.0
26	212.6	213.4	214.0	214.6	215.2	215.8	216.3	217.0	217.6	218.1	218.7	219.3	219.9	220.4	221.0	221.8	222.3
28	204.5	205.0	205.6	206.3	206.9	207.4	208.1	208.7	209.3	209.9	210.5	211.0	211.6	212.2	212.8	213.4	213.9
30	196.9	197.4	198.0	198.5	199.2	199.8	200.3	200.9	201.3	202.0	202.5	203.0	203.6	204.1	204.7	205.3	205.8
32	189.4	190.0	190.6	191.1	191.8	192.4	192.9	193.4	194.0	194.5	195.1	195.6	196.1	196.6	197.2	197.8	198.3
34	182.0	182.6	183.2	183.8	184.3	184.9	185.8	186.0	186.6	187.2	187.6	188.2	188.7	189.3	189.8	190.4	190.9
36	174.6	175.2	175.8	176.4	177.0	177.6	178.2	178.7	179.3	179.8	180.3	180.9	181.4	182.0	182.5	183.0	183.5
38	167.7	168.2	168.8	169.3	170.0	170.5	171.0	171.6	172.2	172.7	173.2	173.8	174.2	174.8	175.3	175.8	176.3
40	160.6	161.1	161.8	162.3	162.9	163.4	164.0	164.5	165.0	165.6	166.1	166.7	167.2	167.8	168.3	168.9	169.4
42	153.9	154.4	155.0	155.6	156.1	156.7	157.3	157.8	158.3	158.8	159.5	160.0	160.4	161.0	161.5	162.0	162.5
44	147.1	147.7	148.2	148.8	149.4	150.0	150.5	151.0	151.5	152.0	152.6	153.1	153.6	154.2	154.7	155.2	155.7
46	140.6	141.1	141.7	142.2	142.8	143.3	143.9	144.5	145.0	145.5	146.0	146.5	147.0	147.6	148.0	148.5	149.0
48	134.0	134.5	135.0	135.6	136.2	136.7	137.2	137.8	138.4	138.9	139.4	140.0	141.5	141.0	141.6	142.1	142.6
50	128.1	128.5	129.1	129.8	130.2	130.9	131.3	131.8	132.4	132.9	133.3	133.9	134.2	134.7	135.2	135.7	136.2

TABLE 18.—PROPERTIES OF SOLUTIONS OF AMMONIA AND WATER.—(Continued.)

Per cent, NH_3.	Absolute pressure.																
	129	130	131	132	133	134	135	136	137	138	139	140	141	142	143	144	145
2	336.7	337.4	338.0	338.7	339.3	339.9	340.5	341.2	341.8	342.4	343.0	343.6	344.2	344.8	345.4	346.0	346.5
4	326.1	326.7	327.4	328.0	328.5	329.2	329.9	330.5	331.1	331.6	332.3	333.0	333.5	334.0	334.6	335.2	335.7
6	315.4	316.0	316.5	317.2	317.9	318.4	319.2	319.8	320.3	321.0	321.6	322.2	322.8	323.4	324.0	324.5	335.0
8	305.2	305.8	306.4	307.0	307.5	308.1	308.7	309.4	309.9	310.5	311.4	311.6	312.3	312.8	313.3	313.9	314.4
10	295.1	295.7	296.3	296.9	297.4	298.2	298.7	299.4	300.0	300.6	301.1	301.8	302.4	302.9	303.4	304.0	304.5
12	285.0	285.6	286.0	286.7	287.3	287.9	288.5	289.2	289.7	290.3	290.9	291.5	292.0	292.5	293.0	293.7	294.3
14	275.5	276.0	276.5	277.2	277.7	278.3	278.9	279.4	280.0	280.6	281.2	281.8	282.3	282.8	283.4	284.0	284.5
16	265.9	266.4	267.0	267.6	268.3	268.8	269.4	270.0	270.6	271.0	271.7	272.2	272.7	273.4	273.8	274.4	274.9
18	256.4	257.0	257.4	258.0	258.6	259.1	259.8	260.3	260.9	261.4	262.0	262.5	263.1	263.6	264.1	264.6	265.3
20	247.3	247.9	248.4	249.0	249.6	250.2	250.7	251.3	251.8	252.4	252.9	253.5	254.0	254.5	255.0	255.6	256.1
22	238.6	239.3	239.8	240.4	241.0	241.5	242.0	242.5	243.1	243.6	244.2	244.7	245.2	245.7	246.2	246.9	247.3
24	230.7	231.3	231.7	232.3	232.9	233.4	233.9	234.5	235.0	235.6	236.1	236.7	237.2	237.7	238.3	238.7	239.3
26	222.8	223.4	224.0	224.5	225.1	225.6	226.0	226.7	227.1	227.6	228.1	228.7	229.2	229.7	230.2	230.7	231.2
28	214.5	215.0	215.5	216.1	216.6	217.3	217.7	218.3	218.8	219.4	219.9	220.4	220.9	221.4	222.0	222.5	223.0
30	206.3	206.9	207.5	208.0	208.5	209.0	209.7	210.2	210.7	211.2	211.7	212.2	212.9	213.2	213.8	214.2	214.7
32	198.8	198.4	199.9	200.4	201.0	201.4	202.0	202.5	203.0	203.5	204.0	204.5	205.0	205.6	206.0	206.5	207.0
34	191.4	191.9	192.4	192.9	103.4	194.0	194.5	195.0	195.5	196.0	196.5	197.0	197.5	198.0	198.5	199.0	199.4
36	184.1	184.7	185.0	185.7	186.2	186.7	187.2	187.6	188.3	188.7	189.3	189.7	190.3	190.7	191.3	191.7	192.2
38	177.0	177.5	178.0	178.5	179.0	179.5	180.0	180.6	181.0	181.5	182.0	182.5	183.0	183.5	184.0	184.5	185.0
40	170.0	170.5	170.9	171.4	172.0	172.5	173.0	173.5	174.0	174.5	175.0	175.5	175.9	176.3	176.8	177.3	177.8
42	163.0	163.5	164.0	164.5	165.0	165.5	166.0	166.5	167.0	167.5	168.0	168.5	169.0	169.5	170.0	170.5	171.0
44	156.2	156.7	157.1	157.7	158.2	158.7	159.1	159.7	160.2	160.7	161.1	161.6	162.0	162.5	163.0	163.5	164.1
46	149.6	150.2	150.5	151.0	151.5	152.0	152.5	153.0	153.5	154.0	154.5	155.0	155.4	155.9	156.4	156.8	157.3
48	143.2	143.7	144.0	144.7	145.2	145.8	146.2	146.7	147.2	147.7	148.1	148.6	149.1	149.6	150.0	150.5	151.0
50	136.8	137.2	137.7	158.2	138.6	139.2	139.6	140.2	140.7	141.1	141.6	142.0	142.5	143.0	143.4	143.8	144.2

TABLE 18.—PROPERTIES OF SOLUTIONS OF AMMONIA AND WATER.—(Continued.)

Per cent, NH_3.	Absolute pressure.																
	146	147	148	149	150	151	152	153	154	155	156	157	158	159	160	161	162
2	347.1																
4	336.4	337.0	337.5														
6	335.7	336.3	336.9	337.4	337.9												
8	315.0	315.5	316.2	316.6	317.3	317.8	318.3										
10	305.0	305.6	306.3	306.8	307.4	307.8	308.4	309.0	309.4	310.0	310.5	311.0	311.6	312.1	312.6	313.0	313.6
12	294.7	295.3	295.8	296.4	297.0	297.5	298.0	298.5	299.0	299.6	300.2	300.7	301.2	301.7	302.2	302.5	303.2
14	285.0	285.6	286.1	286.7	287.3	287.7	288.3	288.8	289.3	289.9	290.4	290.9	291.4	292.0	292.4	292.9	293.3
16	275.5	276.0	276.5	277.0	277.5	278.1	278.6	279.1	279.6	280.3	280.7	281.3	281.7	282.2	282.8	283.1	283.7
18	265.7	266.4	266.9	267.4	268.0	268.5	269.0	269.5	270.0	270.5	271.0	271.4	272.0	273.5	273.0	273.3	273.8
20	256.7	257.3	257.6	258.1	258.6	259.2	259.7	260.3	260.7	261.2	261.7	262.1	262.6	263.2	263.6	264.1	264.5
22	247.9	248.4	248.9	249.4	250.0	250.4	250.9	251.5	251.9	252.4	252.9	253.3	253.9	254.3	254.9	255.3	255.8
24	239.9	240.3	240.8	241.3	241.9	242.3	242.8	243.3	243.7	244.3	244.8	245.3	245.7	246.3	246.7	247.2	247.6
26	231.6	232.3	232.8	233.3	233.7	234.2	234.7	235.1	235.6	236.2	236.6	237.0	237.4	237.9	238.4	238.7	239.3
28	223.5	224.0	224.5	225.0	225.5	226.0	226.5	227.0	227.4	227.9	228.4	228.9	229.3	229.8	230.3	230.7	231.2
30	215.3	215.8	216.3	216.7	217.3	217.7	218.3	218.7	219.2	219.8	220.2	220.7	221.1	221.6	222.0	222.6	223.0
32	207.5	208.0	208.5	209.0	209.4	209.8	210.3	210.7	211.1	211.6	212.1	212.6	213.1	213.5	213.8	214.3	214.6
34	199.9	200.3	200.8	201.3	201.7	202.1	202.6	203.0	203.5	204.0	204.4	204.9	205.3	205.7	206.1	206.5	206.9
36	192.6	193.1	193.6	194.0	194.5	195.1	195.5	196.0	196.4	196.9	197.4	197.8	198.3	198.8	199.1	199.5	199.9
38	185.4	185.9	186.3	186.8	187.3	187.7	188.1	188.6	189.1	189.5	190.0	190.4	190.8	191.2	191.7	192.1	192.6
40	178.2	178.6	179.1	179.6	180.1	180.5	181.0	181.5	182.0	182.4	182.9	183.3	183.8	184.2	184.6	185.0	185.5
42	171.5	172.0	172.4	172.9	173.3	173.8	174.2	174.6	175.1	175.5	175.9	176.3	176.9	177.3	177.7	178.0	178.4
44	164.5	165.0	165.4	165.9	166.3	166.7	167.2	167.5	168.0	168.5	168.7	169.3	169.8	170.3	170.8	171.0	171.4
46	157.7	158.1	158.6	159.0	159.4	159.8	150.3	160.8	161.3	161.7	162.1	162.5	162.9	163.4	163.8	164.2	164.5
48	151.5	152.0	152.4	152.9	153.3	153.7	154.1	154.7	155.0	155.3	155.8	156.3	156.7	157.1	157.5	157.9	158.3
50	144.7	145.1	145.5	146.0	146.4	146.8	147.3	147.7	148.0	148.5	149.0	149.4	149.8	150.3	150.7	151.0	151.5

TABLE 18.—PROPERTIES OF SOLUTIONS OF AMMONIA AND WATER.—(Continued.)

Per cent. NH_3.	Absolute pressure.																
	163	164	165	166	167	168	169	170	171	172	173	174	175	176	177	178	179
2																	
4																	
6																	
8																	
10	314.1	314.7	315.1	315.6	316.1	316.6	317.1	317.6	318.1	318.7	319.2	319.7	320.3	320.8	321.2	321.7	322.2
12	203.7	304.1	304.6	305.1	305.6	306.1	306.6	307.1	307.6	308.1	308.6	309.1	309.6	310.1	310.6	311.0	311.6
14	293.8	294.2	294.8	295.2	295.8	296.2	296.8	297.1	297.6	298.1	298.5	299.0	299.5	300.0	300.5	301.0	301.5
16	284.1	284.6	285.0	285.5	286.0	286.4	286.9	287.3	287.8	288.3	288.7	289.2	289.6	290.1	290.6	291.1	291.5
18	274.3	274.8	275.2	275.7	276.1	276.7	277.2	277.6	278.0	278.5	279.0	279.5	280.0	280.5	281.0	281.4	281.9
20	265.0	265.5	266.0	266.5	267.0	267.5	267.9	268.4	268.9	269.3	269.9	270.3	270.8	271.3	271.8	272.3	272.8
22	256.2	256.6	257.0	257.5	258.0	258.4	258.9	259.3	259.7	260.2	260.6	261.1	261.6	262.1	262.5	263.0	263.4
24	248.0	248.5	249.0	249.4	249.9	250.3	250.6	251.0	251.5	252.0	252.5	253.0	253.4	253.8	254.3	254.6	255.1
26	239.7	240.3	240.7	241.1	241.5	242.0	242.5	243.0	243.4	243.8	244.2	244.6	245.0	245.5	246.0	246.3	246.8
28	231.6	232.0	232.5	232.9	233.3	233.7	234.0	234.5	234.9	235.4	235.8	236.3	236.7	237.1	237.5	238.0	238.5
30	223.5	224.0	224.4	224.9	225.3	225.8	226.3	226.7	227.1	227.5	228.0	228.4	228.8	229.3	229.7	230.1	230.6
32	215.0	215.4	215.9	216.3	216.6	217.0	217.3	217.7	218.2	218.6	219.0	219.5	220.0	220.4	220.8	221.2	221.7
34	207.4	207.7	208.1	208.4	208.8	209.2	209.7	210.0	210.4	210.9	211.3	211.7	212.0	212.5	213.0	213.4	213.8
36	200.3	200.6	201.0	201.4	201.9	202.2	202.5	202.9	203.3	203.7	204.0	204.4	204.8	205.2	205.6	206.0	206.4
38	193.0	193.4	193.8	194.1	194.6	194.9	195.2	195.6	195.9	196.3	196.7	197.0	197.4	197.8	198.2	198.6	199.0
40	186.0	186.4	186.9	187.2	187.7	188.0	188.5	188.9	189.3	189.8	190.2	190.5	190.9	191.2	191.6	192.0	192.4
42	178.9	179.2	179.6	180.0	180.3	180.6	181.0	181.4	181.9	182.2	182.5	183.0	183.3	183.7	184.1	184.5	184.9
44	171.8	172.2	173.5	173.9	173.3	173.6	174.0	174.3	174.7	175.0	175.3	175.8	176.2	176.8	177.0	177.3	177.8
46	165.0	165.3	165.7	166.0	166.4	166.8	167.2	167.5	167.8	168.3	168.6	169.0	169.3	169.7	170.2	170.5	170.9
48	158.7	159.0	159.4	159.8	160.2	160.5	160.7	161.2	161.5	161.9	162.2	162.7	163.0	163.3	163.7	164.1	164.5
50	152.0	152.3	152.7	153.1	153.6	154.1	154.5	154.9	155.2	155.5	156.0	156.3	156.7	157.2	157.6	158.1	158.3

TABLE 18.—PROPERTIES OF SOLUTIONS OF AMMONIA AND WATER.—(Continued.)

Per cent, NH_3.	Absolute pressure.										
	180	181	182	183	184	185	186	187	188	189	190
2											
4											
6											
8											
10	322.7	323.1	323.6	324.0	324.5	325.0	325.5	326.0	326.5	326.0	327.0
12	312.0	312.5	313.0	313.4	313.9	314.3	314.7	315.2	315.7	316.1	316.5
14	302.0	302.4	302.9	303.3	303.7	304.2	304.6	305.1	305.6	306.0	306.5
16	292.0	292.5	293.0	293.5	293.9	294.4	294.9	295.3	295.7	296.2	296.6
18	282.4	282.9	283.3	283.7	284.2	284.7	285.1	285.6	286.0	286.5	287.0
20	273.3	273.6	274.1	274.6	275.0	275.5	276.0	276.5	277.0	277.5	278.0
22	263.9	264.3	264.7	265.2	265.6	266.1	266.5	267.0	267.5	268.0	268.5
24	255.5	256.0	256.5	257.0	257.4	257.8	258.2	258.7	259.1	259.6	260.0
26	247.2	247.6	248.0	248.4	248.9	249.3	249.7	250.2	250.6	251.0	251.5
28	238.9	239.3	239.8	240.2	240.6	241.0	241.5	241.9	242.4	242.8	243.2
30	231.1	231.4	231.8	232.2	232.6	233.1	233.5	234.0	234.4	234.8	235.1
32	222.0	222.5	223.0	223.4	223.8	224.2	224.7	225.1	225.5	226.0	226.4
34	214.2	214.5	215.0	215.4	215.9	216.3	216.7	217.0	217.5	218.0	218.3
36	206.8	207.2	207.5	207.9	208.3	208.7	209.0	209.3	209.8	210.2	210.6
38	199.3	199.7	200.0	200.3	200.7	201.1	201.6	201.9	202.3	202.6	203.0
40	192.8	193.2	193.6	193.9	194.3	194.7	195.0	195.3	195.7	196.1	196.5
42	185.3	185.7	186.0	186.4	186.9	187.3	187.6	188.0	188.3	188.8	189.2
44	178.2	178.5	178.9	179.3	179.7	180.1	180.5	180.9	181.3	181.7	182.0
46	171.3	171.7	172.0	172.4	172.8	173.3	173.6	174.0	174.3	174.7	175.2
48	164.9	165.3	165.8	166.0	166.4	166.7	167.1	167.5	167.9	168.3	168.6
50	158.7	159.1	159.5	159.9	160.3	160.7	161.0	161.4	161.8	162.0	162.4

TABLE 18.—PROPERTIES OF SOLUTIONS OF AMMONIA AND WATER.—(Concluded.)

Per cent. NH_3.	Absolute Pressure.									
	191	192	193	194	195	196	197	198	199	200
2										
4										
6										
8										
10	327.7	328.2	328.5	329.0	329.4	329.9	330.3	330.7	331.1	331.6
12	317.0	317.3	317.9	318.3	318.7	319.1	319.6	320.0	320.0	320.7
14	307.0	307.4	307.9	308.4	308.8	309.3	309.8	310.3	310.7	311.1
16	297.1	297.6	298.0	298.5	299.0	299.5	300.0	300.5	300.9	301.3
18	287.4	287.9	288.4	288.9	289.3	289.9	290.3	290.8	290.3	291.7
20	278.3	278.9	279.3	279.9	280.3	280.8	281.3	281.7	282.1	282.7
22	268.9	269.3	269.8	270.3	270.7	271.2	271.6	272.1	272.5	273.0
24	260.5	260.9	261.4	261.8	262.3	262.7	263.0	263.6	264.0	264.5
26	251.9	252.3	252.8	253.0	253.5	253.9	254.4	254.8	255.3	255.8
28	243.7	244.0	244.5	244.9	245.4	245.8	248.3	246.7	247.2	247.5
30	235.6	236.0	236.4	236.8	237.2	237.7	238.2	238.7	239.0	239.3
32	226.9	227.3	227.7	228.2	228.7	229.1	229.5	230.0	230.4	230.8
34	218.7	219.2	219.6	220.0	221.4	220.8	221.2	221.7	222.0	222.5
36	211.0	211.3	211.6	212.0	212.4	212.9	213.2	213.7	214.0	214.4
38	203.3	203.8	204.2	204.5	204.9	205.2	205.6	206.0	206.4	206.8
40	197.0	197.3	197.7	198.1	198.4	198.7	199.2	199.4	199.9	200.1
42	189.5	190.0	190.3	190.7	191.1	191.5	191.9	192.3	192.7	193.0
44	182.4	182.8	183.2	183.5	183.9	194.3	184.7	185.1	185.5	185.9
46	175.5	175.9	176.3	176.7	177.1	177.4	177.9	178.3	178.7	179.0
48	169.0	169.3	169.7	170.1	170.5	170.9	171.2	171.7	172.0	172.5
50	162.8	163.2	163.5	163.9	164.2	164.7	165.0	165.3	165.6	166.0

TABLE 19.—BOILING TEMPERATURES OF AQUA AMMONIA.

From Practical Refrigerating Engineers' Pocketbook.—Starr.

PER CENT AMMONIA	Pounds Absolute Pressure												
	1	2	3	4	5	6	7	8	9	10	15	20	25
52%	− 57.0	−40.0	−28.0	−19.5	−12.0	− 6.0	+ 1.0	+ 3.5	+ 7.0	+10.0	25.5	39.0	49.0
54%	− 60.5	−44.0	−31.5	−22.5	−16.0	− 9.0	− 4.0	+ 0.5	+ 4.0	+ 7.0	24.5	35.0	45.5
56%	− 63.0	−47.5	−34.5	−25.5	−19.0	−12.5	− 7.0	− 2.5	+ 1.0	+ 4.0	20.5	32.0	42.0
58%	− 65.0	−50.0	−37.0	−28.0	−21.0	−16.0	−10.0	− 5.5	− 2.0	+ 1.0	17.0	29.0	39.0
60%	− 67.0	−52.0	−39.5	−30.5	−23.5	−17.5	−12.5	− 8.5	− 5.0	− 1.5	15.0	26.0	36.0
62%	− 70.0	−55.0	−42.5	−33.5	−26.0	−20.0	−15.0	−11.0	− 7.5	− 4.0	11.5	23.0	33.0
64%	− 72.0	−57.5	−44.5	−36.0	−28.5	−22.5	−18.0	−13.5	−10.0	− 7.0	8.5	20.0	30.0
66%	− 74.0	−60.0	−47.0	−38.5	−31.0	−25.0	−20.0	−16.0	−12.0	− 9.0	7.9	17.5	27.0
68%	− 77.0	−62.5	−50.0	−40.0	−33.0	−27.0	−22.0	−17.5	−14.5	−11.5	+ 4.5	15.0	24.5
70%	− 79.0	−65.0	−51.5	−42.5	−35.0	−29.0	−24.0	−20.5	−17.0	−14.0	+ 2.5	12.5	22.0
72%	− 81.0	−67.0	−53.5	−45.0	−37.0	−31.5	−27.0	−23.0	−19.5	−16.5	− 1.0	10.0	19.0
74%	− 82.5	−69.0	−55.5	−47.0	−39.5	−33.5	−29.0	−25.0	−21.5	−18.5	− 2.5	7.5	17.5
76%	− 84.5	−70.5	−57.5	−49.0	−42.0	−36.0	−31.5	−27.0	−23.5	−20.5	− 5.0	5.5	15.0
78%	− 86.0	−72.5	−59.5	−51.0	−43.5	−38.0	−33.0	−29.0	−25.0	−22.5	− 7.5	+ 3.5	13.0
80%	− 87.5	−74.5	−61.5	−53.0	−45.5	−39.5	−35.0	−31.0	−27.0	−24.5	−10.0	+ 1.0	10.5
82%	− 91.0	−77.0	−63.0	−54.5	−47.0	−41.0	−37.0	−33.0	−30.0	−26.5	−11.5	− 1.0	8.5
84%	− 92.5	−79.5	−65.0	−55.0	−49.0	−42.5	−38.5	−35.0	−32.0	−28.5	−13.0	− 3.0	6.5
86%	− 94.0	−80.5	−67.0	−57.5	−50.5	−44.5	−40.0	−37.5	−34.0	−30.5	−15.0	− 5.0	+ 4.5
88%	− 95.0	−81.5	−69.0	−59.5	−53.0	−47.0	−42.5	−39.0	−35.5	−32.5	−17.0	− 6.5	+ 2.5
90%	− 96.0	−83.0	−71.0	−62.0	−55.0	−49.0	−44.0	−40.5	−37.0	−34.5	−19.0	− 8.0	− 0.5
92%	− 98.0	−84.0	−72.0	−63.5	−56.0	−50.5	−45.5	−42.0	−38.5	−35.5	−20.5	−10.5	− 1.0
94%	− 99.0	−85.0	−74.5	−65.0	−58.0	−52.5	−47.5	−43.5	−40.0	−37.0	−22.0	−12.0	− 3.0
96%	−100.0	−85.5	−75.5	−66.0	−59.5	−54.5	−49.0	−45.0	−42.0	−38.5	−23.5	−13.5	− 4.5
98%	−101.5	−86.0	−77.0	−67.0	−61.0	−55.5	−50.5	−46.0	−43.5	−40.0	−25.5	−14.5	− 6.5
100%	−105.0	−90.0	−80.0	−68.5	−62.5	−57.0	−52.0	−48.5	−45.0	−42.0	−27.5	−16.5	− 8.0

TABLE 19.—BOILING TEMPERATURES OF AQUA AMMONIA.—(Continued.)

Per Cent Ammonia	Pounds Absolute Pressure												
	30	35	40	45	50	55	60	65	70	75	80	85	90
52%	55.5	64.5	70.0	77.0	81.5	87.0	91.6	96.0	100.5	104.5	108.0	111.5	114.5
54%	54.5	61.0	67.0	73.0	78.5	83.5	87.5	92.0	96.5	100.0	104.0	108.0	110.5
56%	50.5	57.5	64.0	70.0	75.0	80.0	84.0	88.5	92.5	97.0	100.0	103.5	106.5
58%	47.0	54.0	60.5	66.5	71.5	76.5	81.0	85.0	89.0	93.0	97.0	100.5	103.0
60%	43.5	51.0	57.5	63.0	66.5	73.0	78.0	82.0	86.0	90.0	93.5	97.0	100.0
62%	41.5	48.0	54.5	60.0	65.0	70.0	75.0	79.0	83.0	87.0	90.5	94.0	97.0
64%	38.0	45.0	51.5	57.0	62.5	67.0	71.5	75.5	80.0	83.0	87.0	91.0	94.0
66%	35.0	42.5	48.0	54.0	59.5	64.0	68.5	72.5	76.5	81.5	84.5	87.5	90.5
68%	33.0	39.5	42.0	51.5	56.5	61.0	65.5	70.0	74.0	77.5	81.0	84.5	87.5
70%	30.0	37.0	43.0	48.0	53.5	58.5	63.0	67.0	71.0	75.0	78.0	81.5	84.5
72%	27.0	34.5	40.5	45.5	51.0	56.0	60.0	64.0	68.5	72.5	75.5	79.0	82.0
74%	25.0	32.0	38.0	43.5	48.5	53.0	57.5	61.5	66.0	69.5	73.0	76.5	79.5
76%	22.5	30.0	36.0	41.5	46.5	51.0	55.5	59.5	63.0	67.0	70.5	74.0	77.0
78%	20.5	27.5	33.0	39.0	44.0	47.5	53.0	57.0	61.0	64.5	68.0	71.5	74.0
80%	18.0	25.0	31.0	36.5	41.5	46.0	51.0	55.0	58.5	62.0	65.5	69.0	71.5
82%	16.0	23.0	29.0	34.5	40.0	44.0	48.5	52.5	56.0	59.5	63.0	66.5	69.5
84%	14.0	21.0	27.0	32.5	37.5	41.5	46.0	50.0	53.5	57.5	61.0	64.0	67.0
86%	12.5	19.0	25.0	30.0	35.0	39.0	44.0	47.5	51.5	55.0	58.0	62.0	65.0
88%	10.0	17.0	23.0	28.5	33.5	37.5	42.0	46.0	49.5	53.5	56.5	60.0	62.5
90%	8.0	15.0	21.0	26.0	31.0	35.0	40.0	43.5	47.5	51.0	54.5	58.0	60.5
92%	6.5	13.0	19.0	24.0	29.0	33.5	37.5	42.0	45.5	49.0	52.0	55.0	58.0
94%	4.5	11.0	17.5	22.5	27.5	31.5	35.1	40.0	43.5	46.5	50.0	53.5	56.5
96%	2.5	9.5	15.5	20.5	25.0	29.5	33.5	38.0	41.5	45.0	48.5	51.5	54.5
98%	+1.0	7.5	13.5	18.5	23.0	27.5	31.5	36.0	40.0	43.0	46.5	50.0	52.5
100%	−0.5	6.0	11.0	17.0	21.5	26.0	30.5	34.5	38.0	41.0	44.5	47.5	50.5

TABLE 19.—BOILING TEMPERATURES OF AQUA AMMONIA.—(Continued)

Per Cent Ammonia	Pounds Absolute Pressure												
	95	100	105	110	115	120	125	130	135	140	145	150	155
52%	118.0	120.5	124.0	126.0	129.5	132.0	134.5	137.0	139.0	141.5	144.0	146.0	148.0
54%	113.0	116.5	120.0	122.5	125.5	128.0	130.5	132.5	135.0	137.5	139.5	142.0	144.0
56%	109.5	113.0	116.0	119.0	121.5	124.0	126.5	128.5	130.5	133.0	135.5	138.0	140.0
58%	106.0	109.5	113.0	115.5	118.5	120.5	123.0	125.0	127.0	129.5	132.0	134.5	136.0
60%	103.0	106.0	109.0	111.5	114.5	116.5	119.0	121.5	123.5	126.0	128.5	130.5	132.0
62%	100.0	103.0	106.0	108.5	111.0	113.5	115.5	118.0	130.5	123.0	125.0	127.0	129.0
64%	96.5	99.5	103.0	105.0	107.5	110.0	112.5	115.0	117.0	119.5	121.5	124.0	125.5
66%	93.5	96.5	99.5	102.0	104.5	107.0	109.5	111.5	113.5	115.5	118.0	120.5	122.0
68%	90.5	93.5	96.5	98.0	101.5	104.0	106.5	108.5	111.0	113.0	114.5	117.0	119.0
70%	87.5	90.5	93.5	96.0	98.5	101.0	103.0	105.5	108.0	110.0	112.0	114.5	116.0
72%	85.0	87.5	91.0	93.0	95.5	98.0	100.5	103.0	105.0	107.0	109.0	111.5	113.0
74%	82.5	85.0	88.0	90.5	93.0	95.5	97.5	100.0	102.5	104.5	106.5	108.5	110.0
76%	79.5	82.0	85.5	87.5	90.5	92.5	95.0	97.5	99.5	101.5	103.5	106.0	107.5
78%	77.0	79.5	83.0	85.0	87.5	90.0	92.0	94.0	97.0	99.0	101.5	103.5	105.0
80%	74.5	77.0	80.5	82.5	85.0	87.5	89.5	92.5	94.5	96.5	98.5	100.5	102.0
82%	72.0	75.0	78.0	80.5	83.0	85.0	87.0	89.5	92.0	94.0	96.0	98.0	99.5
84%	70.0	72.5	75.5	78.0	80.5	83.0	85.0	87.0	89.5	91.5	93.5	95.5	97.5
86%	67.5	70.0	73.5	75.5	78.0	80.5	82.5	85.0	87.0	89.5	91.0	93.0	95.0
88%	65.0	68.0	71.0	73.5	76.0	78.0	80.5	82.5	85.0	87.0	89.0	91.0	92.5
90%	63.0	66.0	69.0	71.0	74.0	76.0	78.5	80.5	82.5	85.0	86.5	88.5	90.5
92%	60.5	64.0	66.5	69.0	71.5	74.0	76.5	78.5	80.5	82.5	84.5	86.5	88.0
94%	59.0	62.0	64.5	66.5	69.5	71.5	74.0	76.5	78.5	80.5	82.5	84.5	86.0
96%	57.5	60.0	63.0	65.0	67.5	70.0	72.5	74.5	76.5	78.5	80.5	82.5	84.0
98%	55.5	58.0	60.5	63.0	65.5	67.5	70.0	72.5	74.5	76.5	78.5	80.5	82.0
100%	53.5	56.0	58.5	61.0	63.5	66.0	68.0	70.5	72.5	74.5	76.5	78.5	80.0

TABLE 19.—BOILING TEMPERATURES OF AQUA AMMONIA.—(Concluded.)

Per Cent Ammonia	Pounds Absolute Pressure											
	160	165	170	175	180	185	190	195	200	205	210	215
52%	150.0	152.5	154.0	157.0	158.5	160.0	161.5	163.5	165.0	167.0	168.5	170.5
54%	146.0	148.0	150.0	152.0	154.0	156.0	157.0	159.5	161.0	162.5	164.5	166.0
56%	142.0	144.0	146.0	148.0	150.0	151.5	153.5	155.0	157.0	158.5	160.0	162.0
58%	138.0	140.5	142.5	144.0	146.0	148.0	149.5	152.5	153.0	154.5	156.0	158.0
60%	134.0	136.5	138.5	140.5	142.5	144.0	146.0	147.5	149.0	151.0	153.0	154.0
62%	131.0	133.0	135.0	137.0	139.0	140.5	142.5	144.0	145.5	147.0	149.0	150.5
64%	127.5	130.0	131.5	133.5	135.5	137.5	139.0	140.5	142.0	144.0	145.5	147.0
66%	124.5	126.5	128.5	130.5	132.0	134.0	135.5	137.0	139.0	140.5	142.0	144.5
68%	121.0	123.0	125.0	127.0	129.0	130.5	132.5	134.0	135.5	137.0	139.0	140.5
70%	118.0	120.0	122.0	124.0	126.0	127.5	129.5	131.0	132.5	134.5	136.5	137.5
72%	115.0	117.0	119.0	121.0	123.0	124.5	126.5	128.0	129.5	131.0	132.5	134.0
74%	112.5	114.5	116.5	118.0	120.0	121.5	123.0	125.0	126.5	128.0	130.0	131.5
76%	110.0	112.0	113.5	115.5	117.5	119.0	120.5	122.0	124.0	125.5	127.0	128.5
78%	107.0	109.0	111.0	112.5	114.5	116.0	118.0	119.5	121.0	122.5	124.0	126.0
80%	104.5	106.5	108.5	110.0	112.0	114.0	115.5	117.0	118.5	120.0	121.5	123.0
82%	102.0	104.0	106.0	107.5	109.0	111.0	112.5	114.0	116.0	117.5	119.0	120.5
84%	99.5	101.5	103.0	105.0	107.0	108.5	110.5	112.0	113.5	115.0	116.5	118.0
86%	97.0	99.0	101.0	102.5	104.5	106.0	107.5	109.5	110.5	112.5	114.0	115.5
88%	95.0	96.5	98.5	100.5	102.0	103.5	105.5	107.0	108.5	110.0	111.5	113.0
90%	92.5	94.5	96.0	98.0	99.5	101.0	103.0	104.5	106.0	107.5	109.0	111.0
92%	90.5	92.0	94.0	95.5	97.5	99.0	101.5	102.5	104.0	105.5	107.0	108.5
94%	87.5	90.0	91.5	93.5	95.5	96.5	98.5	100.0	101.5	103.0	105.0	106.5
96%	86.0	88.0	90.0	91.5	93.0	95.0	96.5	98.0	99.5	101.0	102.5	104.0
98%	84.0	86.0	88.0	89.5	92.0	93.0	94.5	96.0	97.5	99.0	100.5	102.0
100%	82.5	85.5	87.5	89.0	91.0	92.5	94.0	95.5	97.0	97.0	98.5	100.0

TABLE 20.—CALCIUM CHLORIDE BRINE TABLE.
A. S. R. E. Data Book.

Compiled by York Mfg. Co.

Pure CaCl₂ by Weight %	Specific Gravity ‡ 60°F ÷ 39°F	Beaumé Density 60° F.	Specific Heat Btu./lb.° F at 60° F.	Freezing Point °F.	Corresponding *Ammonia Back Pressure† lb./in.² gage	Weight Per Gallon ‡ CaCl₂ lbs.	Weight Per Gallon ‡ Water lbs.	Weight Per Gallon ‡ Brine lbs.	Weight Per Cubic Foot ‡ CaCl₂ lbs.	Weight Per Cubic Foot ‡ Water lbs.	Weight Per Cubic Foot ‡ Brine lbs.
0	1.000	0	1.000	32.0	47.6	0.000	8.35	8.35	0	62.40	62.4
5	1.044	6.1	0.9246	29.0	43.8	0.436	8.281	8.717	3.26	61.89	65.15
6	1.050	7.0	0.9143	28.0	42.6	.526	8.234	8.76	3.93	61.59	65.52
7	1.060	8.2	0.8984	27.0	41.4	.62	8.231	8.851	4.63	61.51	66.14
8	1.069	9.3	0.8842	25.5	39.0	.714	8.212	8.926	5.34	61.36	66.70
9	1.078	10.4	0.8699	24.0	37.9	.81	8.191	9.001	6.05	61.22	67.27
10	1.087	11.6	0.8556	23.0	36.8	0.908	8.168	9.076	6.78	61.05	67.83
11	1.096	12.6	0.8429	21.5	35.0	1.006	8.137	9.143	7.52	60.81	68.33
12	1.105	13.8	0.8284	19.0	32.5	1.107	8.12	9.227	8.27	60.68	68.95
13	1.114	14.8	0.8166	17.0	30.4	1.209	8.093	9.302	9.04	60.47	69.51
14	1.124	15.9	0.8043	14.5	28.0	1.313	8.064	9.377	9.81	60.27	70.08
15	1.133	16.9	0.793	12.5	26.0	1.418	8.034	9.452	10.6	60.04	70.64
16	1.143	18.0	0.7798	9.5	23.4	1.526	8.01	9.536	11.4	59.88	71.26
17	1.152	19.1	0.7672	6.5	20.8	1.635	7.984	9.619	12.22	59.67	71.89
18	1.162	20.2	0.7566	+3.0	18.0	1.747	7.956	9.703	13.05	59.46	72.51
19	1.172	21.3	0.746	0.0	15.7	1.859	7.927	9.786	13.9	59.23	73.13
20	1.182	22.1	0.7375	−3.0	13.6	1.97	7.883	9.853	14.73	58.9	73.63
21	1.192	23.0	0.729	−5.5	11.9	2.085	7.843	9.928	15.58	58.61	74.19
22	1.202	24.4	0.7168	−10.5	8.8	2.208	7.829	10.037	16.50	58.5	75.0
23	1.212	25.5	0.7076	−15.5	5.9	2.328	7.792	10.12	17.4	58.23	75.63
24	1.223	26.4	0.6979	−20.5	3.4	2.451	7.761	10.212	18.32	58.0	76.32
25	1.233	27.4	0.6890	−25.0	1.3	2.574	7.721	10.295	19.24	57.7	76.94
26	1.244	28.3	0.682	−30.0	1.6*	2.699	7.68	10.379	20.17	57.39	77.56
27	1.254	29.3	0.6735	−36.0	6.1*	2.827	7.644	10.471	21.13	57.12	78.25
28	1.265	30.4	0.6657	−43.5	10.6*	2.958	7.605	10.563	22.1	56.84	78.94
29	1.276	31.4	0.6584	−53.0	15.7*	3.09	7.565	10.655	23.09	56.53	79.62
29.5	1.280	31.7	0.6557	−58.0	17.8*	3.153	7.535	10.688	23.56	56.31	79.87

TABLE 21.—SODIUM CHLORIDE (SALT) BRINE TABLE.
A. S. R. E. Data Book.

Compiled by York Manufacturing Co.

Pure NaCl by Weight %	Specific Gravity ‡ 59° F ÷ 39° F.	Beaumé Density † 60° F.	Sal-ometer 60° F ÷ 39° F	Specific Heat 59° Btu. lb. °F	Freezing Point ° F.	Corresponding Ammonia Back Pressure† lb./in.² gage	Weight Per Gallon ‡ NaCl lb./gal.	Weight Per Gallon ‡ Water lb./gal.	Weight Per Gallon ‡ Brine lb./gal.	Weight Per Cubic Foot ‡ NaCl lb./ft.³	Weight Per Cubic Foot ‡ Water lb./ft.³	Weight Per Cubic Foot ‡ Brine lb./ft.³
0	1.000	0.00	0	1.000	32.0	47.6	0.000	8.35	8.35	0.000	62.4	62.4
5	1.035	5.1	18.2	0.938	27.0	41.4	0.432	8.22	8.65	3.230	61.37	64.6
6	1.043	6.1	22.5	0.927	25.5	39.6	0.523	8.19	8.71	3.906	61.19	65.1
7	1.050	7.0	26.0	0.917	24.0	37.9	0.613	8.15	8.76	4.585	60.91	65.5
8	1.057	8.0	29.6	0.907	23.2	37.0	0.706	8.11	8.82	5.280	60.72	66.0
9	1.065	9	33.5	0.897	21.8	35.6	0.800	8.09	8.89	5.985	60.51	66.5
10	1.072	10.10	37.2	0.888	20.4	34.1	0.895	8.05	8.95	6.690	60.21	66.9
11	1.080	10.8	41.1	0.879	18.5	32.0	0.992	8.03	9.02	7.414	59.99	67.4
12	1.087	11.8	44.8	0.870	17.2	30.6	1.090	7.99	9.08	8.136	59.66	67.8
13	1.095	12.7	48.7	0.862	15.5	28.9	1.188	7.95	9.14	8.879	59.42	68.3
14	1.103	13.6	52.6	0.854	13.9	27.4	1.291	7.93	9.22	9.632	59.17	68.8
15	1.111	14.5	56.8	0.847	12.0	25.6	1.392	7.89	9.28	10.395	58.90	69.3
16	1.118	15.4	60.0	0.840	10.2	24.0	1.493	7.84	9.33	11.168	58.63	69.8
17	1.126	16.3	64.0	0.833	8.2	22.3	1.598	7.80	9.40	11.951	58.35	70.3
18	1.134	17.2	68.0	0.826	6.1	20.5	1.705	7.76	9.47	12.744	58.06	70.8
19	1.142	18.1	71.7	0.819	4.0	18.8	1.813	7.73	9.54	13.547	57.75	71.3
20	1.150	19.0	75.2	0.813	+1.8	17.1	1.920	7.68	9.60	14.360	57.44	71.8
21	1.158	19.9	79.1	0.807	−0.8	15.1	2.031	7.64	9.67	15.183	57.12	72.3
22	1.166	20.8	82.8	0.802	−3.0	13.6	2.143	7.60	9.74	16.016	56.78	72.8
23*	1.175	21.7	86.8	0.796	−6.0	11.6	2.256	7.55	9.81	16.854	56.45	73.3
24	1.183	22.5	90.2	0.791	+3.8	18.6	2.371	7.51	9.88	17.712	56.09	73.8
25	1.191	23.4	94.0	0.786	+16.1	29.5	2.488	7.46	9.95	18.575	55.72	74.3
......	1.200	......	......	......	+32.0	47.6	......	......	......	......	......	......

CHAPTER IV.

GENERAL PRINCIPLES OF THE COMPRESSION SYSTEM.

General Principle.—In all mechanical refrigerating machines using a liquefiable fluid the working substance is placed in such a condition that it will absorb heat from a material at a temperature below that of the atmosphere. After the absorption of heat it is placed in such a state that it will give up the absorbed heat and the heat added during the process to a water or air supply at a temperature higher than that of the refrigerator. This is the general principle underlying the operation of refrigerating machines using a liquefiable fluid.

The cycle of operation of the compression system has four principal phases. In the first phase, the refrigerating medium absorbs heat from the material of low temperature. This is accomplished by maintaining a certain pressure upon the medium so that the boiling temperature is a few degrees below the material of low temperature. The heat then flows by natural tendency into the boiling refrigerant, causing it to be entirely evaporated. In the second phase, the vapor of the medium is compressed from the low pressure in the evaporator to a pressure such that the temperature of the condensing medium is a few degrees above the temperature of the available water or air supply, which temperature is always several degrees above that of refrigerator. In the third phase the heat in the medium, that is, the heat absorbed in the evaporator and the heat added during compression, flows by natural tendency into the condenser water or air. The removal of this heat cools and then liquefies the medium. In the fourth phase, the medium is placed in such a condition that it will again absorb heat from the material of low temperature. This is accomplished by reducing the pressure of the liquid of the medium as it leaves the condenser to the pressure existing in the evaporator, by means of the throttle valve, commonly termed expansion valve. The expansion valve reduces the pressure so that the temperature of the boiling fluid is a few degrees below the material of low temperature, while the compressor raises the pressure so that the temperature of the condensing fluid is a few degrees above the temperature of the condenser water.

Action Through the Expansion Valve.—The function of the expansion valve is to throttle the pressure of the refrigerant from the high pressure in the condenser to the low pressure in the evaporating coils and to regulate the flow. The liquid refrigerant is allowed to pass through a small orifice in the expansion or throttle valve. The throttling through the expansion valve takes place without the addition or extraction of heat. Therefore, it is evident that the heat content of the ammonia remains constant in passing through the expansion valve (provided that the velocity of the ammonia is the same before and after passing through the valve).

Evaporation of the Liquid Refrigerant.—If, in a refrigeration plant, it is desired to maintain the temperature of a cold room at 25° F. the pressure on the boiling refrigerant in the refrigerating coil may be held at such a point that the temperature of the boiling refrigerant is 5° F. The heat will then flow from the room into the evaporating refrigerant. The exact amount of heat absorbed by each pound of refrigerant will, of course, vary with each individual refrigerant. The theory underlying the operation of the compression systems is independent of the refrigerants which are used, and will hold true for any liquefiable refrigerant such as ammonia, Freon, or carbon dioxide. Since ammonia is used most extensively in the United States, the various heat quantities, etc., will be calculated for ammonia.

Thus the temperature of the boiling ammonia in the refrigerating coil will be 5° F. when the pressure is maintained at 34.27 lbs. abs. The exact amount of heat that may be absorbed by each pound of the ammonia under the above condition will depend upon the temperature of the liquid before it is introduced into the evaporating coil. If the temperature of the pound of the liquid is 5° F., the heat absorbed will be equal to the latent heat of evaporation, which in this case is 565 Btu.

However, in most cases the temperature of the liquid as it enters the evaporating coil is near the temperature of the liquid as it leaves the condenser. This temperature, in general, is several degrees above the temperature of the boiling ammonia in the refrigerator, so that part of the liquid ammonia must be evaporated in order to cool the remainder to the temperature desired in the refrigerator coil. Thus, at a point just before the expansion valve a pound of the ammonia is all liquid, while immediately after the expansion valve the pound of ammonia is a mixture of liquid and vapor. As an example it may be assumed that the temperature of the liquid ammonia is 86° F. as it enters the evaporating coil. The heat required to cool the liquid ammonia from 86° to 5° F. may be found by multiplying together the temperature range and the average specific heat of the liquid ammonia, or it may be found by simply subtracting the heat content of the liquid at

5° F. from the heat content of the liquid at 86° F. This is shown as follows:

Heat content of liquid at 86° F	138.9 Btu.
Heat content of liquid at 5° F	48.3 Btu.
Total heat to cool liquid	90.6 Btu.

If, by means of the use of cold water, it is possible to cool the liquid ammonia to 66° F., the heat required to cool the liquid to 5° F. is found as follows:

Heat content of liquid at 66° F	116.0 Btu.
Heat content of liquid at 5° F	48.3 Btu.
Total heat to cool liquid	67.7 Btu.

It is now evident that the cooling effect due to the latent heat of evaporation will be reduced by the above amounts of heats required to cool the liquid to temperature of the refrigerator. Thus, the heat available for refrigeration of the room, water, brine, etc., would be as follows:

For 86° Liquid Ammonia:
Cooling effect $= 565.0 - 90.6 = 474.4$ Btu.

For 66° Liquid Ammonia:
Cooling effect $= 565.0 - 67.7 = 497.3$ Btu.

The relative magnitudes of the cooling effects, 474.4 and 497.3 Btu., indicate the desirability of cooling the liquid refrigerant to as low a temperature as economical. This cooling should be accomplished by the use of cool water or other means, and not by means of the return vapor from the refrigerating coil.

The percentage of the ammonia that is evaporated in passing through the expansion valve may be calculated readily, as shown below.

This is in reality the quality of the mixture after it has passed the expansion valve, and it represents the loss of refrigerating effect due to above cause.

For 86° Liquid Ammonia:

Quality .... $\frac{90.6}{565.0} = 0.160 = 16\%$

For 66° Liquid Ammonia:

Quality .... $\frac{67.7}{565.0} = 0.120 = 12\%$

Refrigerating Effect.—From the foregoing it will be noted that the available amount of heat which may be used for refrigeration depends

upon the latent heat of evaporation and the sensible heats required to cool the liquids, and in the above cases the refrigerating effects were equal to 474.4 and 497.3 Btu. respectively. The refrigerating effect in general is the amount of heat absorbed by the refrigerant in passing through the refrigerator and is represented by the difference of the heat contents of the refrigerant entering and leaving the evaporating coil. It will also be remembered that the action through the expansion valve is one of constant heat content; that is, the heat content of the liquid before the expansion is the same as the heat content of the mixture after passing through the valve. Therefore, it is evident that the refrigerating effect is equal to the heat content of the saturated vapor leaving the evaporator coils minus the heat content of the liquid refrigerant just before the expansion valve. The refrigerating effects of the ammonia under the conditions in the foregoing examples may be calculated as follows:

For 86° Liquid Ammonia:

Heat content of saturated ammonia vapor at 5° F	613.3 Btu.
Heat content of liquid ammonia at 86° F	138.9 Btu.
Refrigerating effect	474.4 Btu.

For 66° Liquid Ammonia:

Heat content of saturated ammonia vapor at 5° F	613.3 Btu.
Heat content of liquid ammonia at 66° F	116.0 Btu.
Refrigerating effect	497.3 Btu.

It will be noted that this method gives the same results as were obtained in the previous calculations.

These two methods of determining the net refrigerating effect of the ammonia may be stated symbolically as follows:

$$R = l - (h_2 - h_1)$$

where R = refrigerating effect in Btu. per lb. of ammonia

l = latent heat of vaporization in Btu. per lb. ammonia

h_2 = total heat of the liquid at temperature just before the expansion valve

h_1 = heat content of the liquid at the temperature existing in the evaporator

also, $R = H_1 - h_2$

where H_1 = heat content of vapor leaving refrigerator coils, Btu. per lb. of ammonia

The refrigerating effects per pound of ammonia for the usual pressures and temperatures are indicated in Table 22.

In comparing the values of various factors shown in Tables 22, 24, 25, 28, 30, 31 with the results obtained by the equations, it will be noted that there are some slight differences. This is due to the fact that the values shown in the tables were calculated from the original Bureau of Standards Ammonia Tables, while the values illustrated

in the various examples were taken from the later Bureau of Standards Tables of the Properties of Ammonia.

Amount of Fluid to be Evaporated.—The amount of refrigerant to be evaporated in one minute to absorb heat at the rate of one ton of refrigeration depends upon the refrigerating effect of a pound of the refrigerant under the desired conditions of pressures and temperatures. In general, the refrigerating effect is taken to be the difference of the heat content of the vapor leaving the coil and the heat content of the liquid just before the expansion valve. Under ordinary conditions the heat content of the vapor is taken as that of the saturated vapor at the temperature corresponding to the evaporator pressure and the heat content of the liquid is taken at the temperature of saturation due to the condenser pressure.

In the previous examples the refrigerating effects of 474.4 and 497.3 Btu. per pound were obtained. Now, since one ton of refrigeration is equivalent to the removal of heat at the rate of 200 Btu. per minute, it is necessary to evaporate only

$$\frac{200}{474.4} = 0.4216 \text{ lbs. and } \frac{200}{497.3} = 0.4020 \text{ lbs.}$$

per minute of ammonia to produce one ton of refrigeration. The amount of refrigerant circulated by a machine could be found by multiplying together its capacity in tons and the pounds of the refrigerant evaporated per minute per ton. The weight of refrigerant circulated will depend upon the suction and condenser pressures, which in turn determine the refrigerating effect.

This may be expressed as follows:

$$M = \frac{200}{R}$$

where M = lb. per min. per ton.

For standard conditions, the amount is:

$$M = \frac{200}{474.4} = 0.4216 \text{ lbs.}$$

The variation of the weight of ammonia to be evaporated per minute to produce one ton of refrigeration under usual pressures is shown by Table 23.

Compressor Cylinder Displacement.—The piston displacement of a compressor cylinder that is necessary to produce a given amount of refrigeration depends upon the suction and condensing pressures, together with the displacement or volumetric efficiency of the com-

TABLE 22.—REFRIGERATING EFFECT OF AMMONIA, BTU. PER LB.

(Dry Compression with No Liquid Sub-Cooling.)

Gauge Pressure Lbs. per sq. in.	Saturated Evaporating Temperature	Gauge Pressures, lbs. per sq. in. 92.9	103.1	114.1	125.8	138.3	151.7	154.5	165.9	181.1	197.2	214.2	232.3
		Saturated Condensing Temperatures 60°F.	65°	70°	75°	80°	85°	86°	90°	95°	100°	105°	110°
8.7″	−40°F.	487.8	482.2	476.5	470.8	465.0	459.3	458.1	453.5	447.7	441.8	435.9	430.0
5.4″	−35°	489.7	484.1	478.4	472.7	466.9	461.2	460.0	455.4	449.6	443.7	437.8	431.9
1.6″	−30°	491.7	486.1	480.4	474.7	468.9	463.2	462.0	457.4	451.6	445.7	439.8	433.9
1.3	−25°	493.6	488.0	482.3	476.6	470.8	465.1	463.9	459.3	453.5	447.6	441.7	435.8
3.6	−20°	495.5	489.9	484.2	478.5	472.7	467.0	465.8	461.2	455.4	449.5	443.6	437.7
6.2	−15°	497.2	491.6	485.9	480.2	474.4	468.7	467.5	462.9	457.1	451.2	445.3	439.4
9.0	−10°	499.0	493.4	487.7	482.0	476.2	470.5	469.3	464.7	458.9	453.0	447.1	441.2
12.2	− 5°	500.7	495.1	489.4	483.7	477.9	472.2	471.0	466.4	460.6	454.7	448.8	442.9
15.7	0°	502.4	496.8	491.1	485.4	479.6	473.9	472.7	468.1	462.3	456.4	450.0	444.6
19.6	5°	504.0	498.4	492.7	487.0	481.2	475.5	474.3	469.7	463.9	458.0	451.6	446.2
23.8	10°	505.6	500.0	494.3	488.6	482.8	477.1	475.9	471.3	465.5	459.6	453.2	447.8
28.4	15°	507.1	501.5	495.8	490.1	484.3	478.6	477.4	472.8	467.0	461.1	454.7	449.3
33.5	20°	508.6	503.0	497.3	491.6	485.8	480.1	478.9	474.3	468.5	462.6	456.2	450.8
39.0	25°	510.1	504.5	498.8	493.1	487.3	481.6	480.4	475.8	470.0	464.1	457.7	452.3
45.0	30°	511.4	505.8	500.1	494.4	488.6	482.9	481.7	477.1	471.3	465.4	459.0	453.6
51.6	35°	512.7	507.1	501.4	495.7	489.9	484.2	473.0	478.4	472.6	466.6	460.3	454.9
58.6	40°	514.1	508.5	502.8	497.1	491.3	485.6	484.4	479.8	474.0	468.1	461.7	456.3

″ = Inches of mercury below one standard atmosphere (29.92 in.)

pressor cylinder. The pounds of refrigerant to be evaporated per minute per ton of refrigeration is determined by the heat content of the vapor leaving the evaporator and the heat content of the liquid just before the expansion valve, as indicated previously. Now, as soon as the weight of fluid to be evaporated is known, the theoretical amount of displacement in the compressor cylinder can be determined and will be equivalent to the volume of the necessary weight of fluid. The compressor must draw into its cylinder during the suction strokes this volume of refrigerant in order to produce the refrigeration at the desired rate. On the compression strokes the trapped refrigerant is discharged to the condenser.

As an example in the above theory, the displacement required in the cylinder of an ammonia compressor per minute per ton of refrigeration will be considered, assuming a saturated suction temperature of 0° F. and a saturated condensing temperature of 86° F. These temperatures correspond approximately to the pressures of 30 and 170 lbs. per sq. in. abs., approximately, in the evaporator and the condenser.

The refrigerating effect of a pound of ammonia is found as follows:

Heat content of saturated vapor at 30 lbs.	611.6 Btu.
Heat content of liquid at 170 lbs.	139.3
Refrigerating effect per lb.	472.3 Btu.

The amount of ammonia to be evaporated per minute per ton of refrigeration in pounds is calculated as follows:

$$200 \div 472.3 = 0.423$$

It is evident that the compressor must remove the above amount of ammonia from the evaporator in one minute. In order to do this, the compressor cylinder must have a displacement which is at least equivalent to the volume of the above amount at the temperature and pressure of the evaporator. The volume of one pound of ammonia of 0° F. temperature and 30 lbs. pressure is 9.236 cu. ft.

Hence the necessary displacement in this case is calculated as follows:

$$9.236 \times 0.423 = 3.91 \text{ cu. ft. per min.}$$
$$\text{or } 3.91 \times 1728 = 6750.0 \text{ cu. in. per min.}$$

The quantity 6750.0 is the theoretical number of cubic inches of saturated vapor which must be removed from the evaporator in order to produce a ton of refrigeration under the given conditions. However, the compressor cylinder is never able to draw in the above weight from the evaporator on account of inherent defects. Thus, since the weight of the vapor in the cylinder is less than the amount apparently displaced by the piston, the actual displacement per unit of refrigeration must be larger than the above theoretical amount.

TABLE 23.—POUNDS OF AMMONIA PER MINUTE PER TON OF REFRIGERATION.

(Dry Compression with No Liquid Sub-Cooling.)

Gauge Pressure Lbs. per sq. in.	Saturated Evaporating Temperature	Gauge Pressures, lbs. per sq. in. 92.9	103.1	114.1	125.8	138.3	151.7	154.5	165.9	181.1	197.2	214.2	232.3
		Saturated Condensing Temperatures 60°F.	65°	70°	75°	80°	85°	86°	90°	95°	100°	105°	110°
8.7*	−40°F.	0.410	0.414	0.419	0.425	0.431	0.435	0.437	0.442	0.447	0.452	0.459	0.465
5.4*	−35°	0.409	0.413	0.418	0.423	0.428	0.433	0.435	0.439	0.445	0.451	0.457	0.463
1.6*	−30°	0.407	0.411	0.416	0.422	0.426	0.432	0.434	0.437	0.443	0.449	0.455	0.462
1.3	−25°	0.406	0.410	0.414	0.419	0.424	0.430	0.432	0.436	0.441	0.447	0.453	0.459
3.6	−20°	0.404	0.408	0.413	0.418	0.423	0.428	0.429	0.434	0.438	0.445	0.451	0.457
6.2	−15°	0.402	0.407	0.412	0.416	0.422	0.427	0.428	0.433	0.437	0.443	0.449	0.455
9.0	−10°	0.401	0.406	0.410	0.415	0.420	0.425	0.427	0.432	0.436	0.442	0.447	0.453
12.2	− 5°	0.399	0.404	0.409	0.414	0.418	0.423	0.425	0.429	0.434	0.441	0.446	0.452
15.7	0°	0.398	0.402	0.407	0.412	0.417	0.422	0.424	0.427	0.433	0.438	0.444	0.450
19.6	5°	0.397	0.401	0.406	0.411	0.415	0.421	0.422	0.426	0.432	0.436	0.443	0.448
23.8	10°	0.395	0.400	0.405	0.409	0.414	0.419	0.420	0.424	0.429	0.435	0.442	0.447
28.4	15°	0.394	0.399	0.404	0.408	0.413	0.418	0.419	0.423	0.428	0.434	0.441	0.446
33.5	20°	0.393	0.398	0.403	0.407	0.412	0.417	0.418	0.422	0.427	0.433	0.438	0.444
39.0	25°	0.392	0.397	0.402	0.406	0.411	0.415	0.416	0.321	0.426	0.432	0.436	0.442
45.0	30°	0.391	0.396	0.400	0.405	0.410	0.414	0.415	0.419	0.424	0.429	0.435	0.441
51.6	35°	0.390	0.395	0.399	0.403	0.409	0.413	0.414	0.418	0.423	0.428	0.434	0.440
58.6	40°	0.389	0.394	0.398	0.402	0.407	0.412	0.413	0.417	0.422	0.427	0.433	0.439

* = Inches of mercury below one standard atmosphere (29.92 in.).

The theoretical displacement of the compressor may be calculated as follows:

$$T.D. = \frac{200V}{H_1 - h_2}$$

in which $T.D.$ = theoretical displacement cu. ft. per ton of refrigeration.

V = specific volume of refrigerant, cu. ft.

The foregoing formula is based upon the assumption that the throttling process in the expansion valve occurs at constant heat of the refrigerant, that is, the refrigerant does not absorb heat from the outside, but the heats of liquid and evaporation may interchange to a certain extent.

To facilitate the calculation of numerical examples, it will be assumed further that the vapor leaves the evaporator in the dry and saturated state and that the liquid reaches the expansion valve at the saturated temperature due to the pressure in the condenser.

For ammonia, at the standards conditions of 5° F. and 86° F., saturated temperatures in the evaporator and condenser, respectively, the theoretical displacement may be calculated as follows:

$$T.D. = \frac{200 \times 8.150}{613.3 - 138.9} = 3.436 \text{ cu. ft. per ton of refrigeration per min.}$$

The theoretical cylinder displacement in cubic inches per ton of refrigeration may be found in Table 24.

Volumetric Efficiency.—The volumetric or displacement efficiency of a compressor cylinder is the ratio of the volume of the refrigerant actually removed, reduced to the conditions of temperature and pressure of the evaporator, to the volume of the piston displacement.

Several factors affect the amount of the vapor that actually gets into the cylinder. The weight of vapor that would fill the volume of the piston displacement at the condition of the evaporator is always greater than the amount actually displaced or removed. The vapor upon entering the cylinder becomes superheated due to wire-drawing of the gas through the valves and the exposure to the hot cylinder walls, piston and valves. Being at a temperature several degrees above that of the evaporator at the end of the suction stroke, it is naturally less dense, and weighs less per unit volume, since the change of density of a vapor or gas is approximately proportional to the change of the absolute temperature.

There is also a reduction of pressure in the cylinder below that of the evaporator in order to cause the refrigerant to flow through the more or less restricted area of the parts and valves, at a high velocity.

TABLE 24.—THEORETICAL DISPLACEMENT OF AMMONIA COMPRESSORS, CU. IN. PER MIN. PER TON OF REFRIGERATION

(Dry Compression with No Liquid Sub-Cooling.)

Gauge Pressure Lbs. per sq. in.	Saturated Evaporating Temperature	Gauge Pressures, lbs. per sq. in.											
		92.9	103.1	114.1	125.8	138.3	151.7	154.5	165.9	181.1	197.2	214.2	232.3
		Saturated Condensing Temperatures											
		60°F.	65°	70°	75°	80°	85°	86°	90°	95°	100°	105°	110°
8.7*	−40°F.	17610	17770	17790	18150	18520	18630	18770	18870	19080	19450	19540	19960
5.4*	−35°	15320	15460	15650	15830	16020	16220	16280	16430	16620	16880	17020	17370
1.6*	−30°	13340	13400	13600	13830	13950	14160	14220	14300	14520	14710	14920	15140
1.3	−25°	11680	11790	11910	12050	12190	12370	12430	12450	12690	12860	13030	13210
3.6	−20°	10240	10350	10470	10610	10720	10650	10870	11010	11120	11280	11450	11580
6.2	−15°	9015	9122	9240	9323	9460	9560	9580	9700	9800	9955	10060	10220
9.0	−10°	7972	8063	8150	8250	8345	8445	8483	8582	8683	8790	8880	9001
12.2	− 5°	7050	7140	7228	7318	7385	7455	7510	7580	7665	7790	7880	7981
15.7	0°	6268	6344	6422	6490	6565	6646	6680	6723	6819	6899	6992	7083
19.6	5°	5590	5643	5705	5784	5850	5925	5942	5999	6081	6140	6216	6319
23.8	10°	4990	5050	5113	5162	5230	5292	5305	5353	5420	5493	5582	5646
28.4	15°	4463	4520	4589	4628	4685	4740	4754	4800	4853	4925	5002	5058
33.5	20°	4013	4066	4120	4158	4219	4260	4270	4312	4363	4421	4473	4535
39.0	25°	3617	3660	3709	3744	3793	3828	3838	3881	3931	3985	4020	4091
45.0	30°	3261	3290	3335	3368	3419	3453	3462	3496	3535	3578	3628	3678
51.6	35°	2950	2985	3017	3045	3092	3105	3115	3160	3199	3236	3262	3325
58.6	40°	2672	2708	2733	2762	2797	2830	2838	2863	2900	2916	2980	3017

* = Inches of mercury below one standard atmosphere (29.92 in.).

This further lowers the density of the vapor. Also there might be leakage of gas past the piston and valves, which would prevent some vapor from being removed from the evaporator, by occupying space in the effective volume of the cylinder. Again the gases which are compressed into the clearance spaces will re-expand down to the evaporator pressure before the suction valves may open. It is obvious that all of the foregoing factors reduce the weight of vapor that may actually get into the cylinder.

The amount of linear clearance in the cylinders of vertical single-acting ammonia compressors is usually the smallest amount which may be used without allowing the parts to come into direct contact. This will usually amount to $\frac{1}{64}$ to $\frac{1}{32}$ of an inch. In respect to the volumetric efficiency due to the superheat of the vapor during the suction stroke, it will be observed that there are a number of factors which affect the relative magnitude of this efficiency. In consideration of this fact, it is therefore practically impossible to determine these efficiencies by mathematical analysis. These efficiencies due to superheating may, therefore, be determined only by actual tests on an actual ammonia compressor cylinder, and should therefore be termed an operating characteristic of the compressor cylinder.

For the purpose of comparison and calculation, the author has used tests made on actual ammonia compressor cylinders for the purpose of determining the relative magnitude of the efficiency due to superheating. Data taken from the most authoritative tests performed up to date were plotted on curves from which, by means of extension and extrapolation, the values of the volumetric efficiencies for the assumed conditions were obtained. The values of the volumetric efficiencies due to superheating, as determined by this means for vertical single-acting ammonia compressors are shown by Table 25. The values shown by Table 25 are the volumetric efficiencies with a factor of safety of approximately ten per cent.

As previously indicated, these may be termed volumetric efficiencies due to superheating, because of the fact that a very large proportion of the loss of efficiency is due to the superheating effect. The tests of the actual refrigerating machines were conducted by experts, after all parts of the system had been put into as nearly perfect mechanical condition as possible, clearances reduced to the minimum, etc. Therefore, the values of the volumetric efficiencies represent primarily the effect of superheating. It is well to point out that it is possible to obtain volumetric efficiencies which are better than those shown by Table 25, but that these values may be assumed to represent good practice.

Similar values for volumetric efficiencies of horizontal double-acting compressors are shown in Table 26.

TABLE 25.—VOLUMETRIC EFFICIENCIES DUE TO SUPERHEATING FOR VERTICAL SINGLE-ACTING COMPRESSORS.

Gauge Pressure Lbs. per sq. in.	Saturated Evaporating Temperature	Gauge Pressures, lbs. per sq. in. 92.9	103.1	114.1	125.8	138.3	151.7	154.5	165.9	181.1	197.2	214.2	232.3
		Saturated Condensing Temperatures 60°F.	65°	70°	75°	80°	85°	86°	90°	95°	100°	105°	110°
8.7″	−40°F.	76.5	76.0	75.2	74.5	73.5	72.5	72.3	71.4	70.2	69.1	67.8	66.3
5.4″	−35°	77.2	77.3	76.6	75.8	74.9	73.9	73.7	72.7	71.5	70.5	69.2	67.7
1.6″	−30°	79.5	78.7	77.9	77.2	76.3	75.3	75.0	74.2	72.9	71.8	70.4	69.0
1.3	−25°	80.8	80.0	79.3	78.4	77.6	76.6	76.3	75.4	74.3	73.2	71.8	70.3
3.6	−20°	82.2	81.4	80.6	79.7	78.8	78.0	77.6	76.7	75.6	74.4	73.5	71.7
6.2	−15°	83.5	82.7	81.9	81.0	80.2	79.1	78.8	78.0	76.8	75.7	74.3	73.0
9.0	−10°	84.7	84.0	83.1	82.2	81.3	80.4	80.1	78.3	78.0	76.9	75.6	74.2
12.2	− 5°	85.8	85.1	84.3	83.3	82.5	81.5	81.2	80.4	79.3	78.1	76.7	75.3
15.7	0°	86.9	86.3	85.3	84.4	83.6	82.6	82.3	81.5	80.3	79.2	77.8	76.3
19.6	5°	88.0	87.4	86.4	85.6	84.7	83.7	83.4	82.5	81.4	80.3	78.8	77.3
23.8	10°	89.0	88.4	87.5	86.6	85.7	84.7	84.4	83.5	82.4	81.3	79.8	78.3
78.4	15°	90.0	89.3	88.3	87.4	86.6	85.6	85.3	84.4	83.2	82.2	80.7	79.2
33.5	20°	90.9	90.2	89.2	88.3	87.3	86.4	86.1	85.2	83.9	82.7	81.3	79.8
39.0	25°	91.7	90.7	89.8	88.9	87.8	86.8	86.6	85.7	84.4	83.2	81.7	80.4
45.0	30°	92.4	91.4	90.5	89.6	88.5	87.3	87.0	86.2	84.8	83.6	82.0	80.6
51.6	35°	93.2	92.1	91.1	90.1	88.8	87.6	87.3	86.4	85.2	83.7	82.3	80.9
58.6	40°	93.8	92.6	91.5	90.6	89.3	87.8	87.5	86.6	85.3	83.8	82.4	81.1

″ = Inches of mercury below one standard atmosphere (29.92 in.).

TABLE 26.—VOLUMETRIC EFFICIENCIES OF HORIZONTAL DOUBLE-ACTING AMMONIA COMPRESSORS.

Evaporating Gauge Pressure	Saturated Evaporating Temperature	CONDENSER GAUGE PRESSURE											
		92.9	103.1	114.1	125.8	138.3	151.7	154.5	165.9	181.1	197.2	214.2	232.3
		SATURATED CONDENSING TEMPERATURES											
		60° F.	65°	70°	75°	80°	85°	86°	90°	95°	100°	105°	110°
8.7"	—40° F.	64.9	64.4	63.9	62.9	61.9	60.9	60.7	59.8	58.6	57.5	56.2	54.7
5.4"	—35°	65.8	65.7	65.1	64.3	63.4	62.4	62.2	61.2	60.0	59.0	57.7	56.2
1.6"	—30°	68.2	67.5	66.6	65.9	65.0	64.0	63.7	62.9	61.6	60.5	59.1	57.7
1.3 lbs.	—25°	69.6	68.8	68.1	67.2	66.4	65.4	65.1	64.2	63.1	62.0	60.6	59.1
3.6 lbs.	—20°	71.2	70.4	69.6	68.7	67.8	67.0	66.6	65.7	64.6	63.4	62.5	60.7
6.2 lbs.	—15°	72.6	71.8	71.0	70.1	69.3	68.2	67.9	67.1	65.9	64.8	63.5	62.1
9.0 lbs.	—10°	74.1	73.4	72.5	71.6	70.7	69.8	69.5	67.7	67.4	66.3	65.0	63.6
12.2 lbs.	— 5°	75.5	74.8	74.0	73.0	72.2	71.2	70.9	70.1	69.0	67.8	66.4	65.0
15.7 lbs.	0°	76.9	76.3	75.3	74.4	73.6	72.6	72.3	71.5	70.3	69.2	67.8	66.3
19.6 lbs.	5°	78.4	77.8	76.8	76.0	75.1	74.1	73.8	72.9	71.8	70.7	69.2	67.7
23.8 lbs.	10°	80.0	79.4	78.5	77.6	76.7	75.7	75.4	74.5	73.4	72.3	70.8	69.3
28.4 lbs.	15°	81.7	81.0	81.0	79.1	78.3	77.3	77.0	76.1	74.9	73.9	72.4	70.9
33.5 lbs.	20°	83.5	82.8	81.8	80.9	79.9	79.0	78.7	77.8	76.5	75.3	73.9	72.4
39.0 lbs.	25°	85.2	84.2	83.3	82.4	81.3	80.3	80.1	79.2	77.9	76.7	75.2	73.9
45.0 lbs.	30°	86.8	85.8	84.9	84.0	82.9	81.7	81.4	80.6	79.2	78.0	76.4	75.0
51.6 lbs.	35°	88.4	87.3	86.3	85.3	84.0	82.8	82.5	81.6	80.4	78.9	77.5	76.1
58.6 lbs.	40°	89.6	88.4	87.3	86.4	85.1	83.6	83.3	82.4	81.1	79.6	78.2	76.9

" = Inches of mercury.

The effect of clearance upon the volumetric efficiency of a compressor cylinder may be determined by mathematics, as soon as the amount of clearance is known. The thermodynamical expression for the volumetric efficiency due to clearance is shown as follows:

$$E_c = 1 + c - c\left(\frac{(p_2)}{(p_1)}\right)^{\frac{1}{n}}$$

$$\text{or } E_c = 1 - c\left[\left(\frac{(p_2)}{(p_1)}\right)^{\frac{1}{n}} - 1\right]$$

where E_c = volumetric efficiency due to clearance
c = percentage of clearance in per cent of cylinder volume
p_1 = absolute evaporator or suction pressure
p_2 = absolute condenser pressure
n = exponent in the compression law, PV^n = a constant

The value of the exponent, n, in the characteristic law of compression, PV^n = a constant, will vary within certain limits, due to the fact that the ammonia vapor is not a perfect gas. However, the latest data from the Bureau of Standards on the properties of ammonia vapor seem to indicate that the average value for n may be taken to be 1.28. Introducing the value, $n = 1.28$, in the foregoing expression, leads to the following formula for the volumetric efficiency due to clearance:

$$E_c = 1 - c\left[\left(\frac{(p_2)}{(p_1)}\right)^{0.7812} - 1\right]$$

The foregoing expression may be further simplified by the following:

$$F = \left[\left(\frac{(p_2)}{(p_1)}\right)^{0.7812} - 1\right]$$

$$\text{Thus } E_c = 1 - c \times F$$

Using the pressures corresponding to the temperatures under standard conditions, the values of the factor, F, for those conditions may be determined as follows:

$$F = \left[\left(\frac{(169.2)}{(34.27)}\right)^{0.7812} - 1\right] = (3.48 - 1) = 2.48$$

If the amount of clearance is assumed to be 3 per cent of the cylinder volume, the efficiency due to clearance in this case would be found as follows:

$$E_c = 1 - 0.03 \times 2.48 = 0.9256 = 92.56\%$$

TABLE 27.—FACTORS FOR CALCULATING VOLUMETRIC EFFICIENCIES DUE TO CLEARANCE.

Gauge Pressure Lbs. per sq. in.	Saturated Evaporating Temperature	Gauge Pressures, lbs. per sq. in. 92.9	103.1	114.1	125.8	138.3	151.7	154.5	165.9	181.1	197.2	214.2	232.3
		Saturated Condensing Temperatures 60°F.	65°	70°	75°	80°	85°	86°	90°	95°	100°	105°	110°
8.7*	−40°F.	5.190	5.650	6.130	6.640	7.170	7.710	7.820	8.290	8.890	9.065	10.080	10.850
5.4*	−35°	4.530	4.940	5.360	5.810	6.280	6.780	6.870	7.290	7.820	8.380	8.960	9.058
1.6*	−30°	3.940	4.360	4.690	5.070	5.520	5.950	6.030	6.310	6.890	7.390	7.910	8.460
1.3	−25°	3.434	3.760	4.100	4.463	4.841	5.230	5.310	5.640	6.061	6.520	6.990	7.490
3.6	−20°	2.830	3.282	3.585	3.916	4.260	4.610	4.680	4.980	5.370	5.770	6.180	6.640
6.2	−15°	2.598	2.860	3.140	3.438	3.740	4.060	4.120	4.390	4.743	5.074	5.480	5.890
9.0	−10°	2.256	2.493	2.742	3.008	3.288	3.565	3.640	3.880	4.190	4.520	4.865	5.220
12.2	− 5°	1.950	2.168	2.394	2.630	2.890	3.145	3.198	3.420	3.716	4.010	4.320	4.640
15.7	0°	1.684	1.940	2.088	2.304	2.534	2.771	2.820	3.022	3.280	3.550	3.813	4.135
19.6	5°	1.443	1.625	1.812	2.080	2.232	2.432	2.480	2.663	2.897	3.150	3.403	3.671
23.8	10°	1.232	1.395	1.568	1.748	1.936	2.137	2.166	2.344	2.560	2.784	3.018	3.270
28.4	15°	1.042	1.183	1.348	1.516	1.680	1.868	1.907	2.002	2.257	2.408	2.676	2.902
33.5	20°	0.872	1.090	1.155	1.306	1.466	1.632	1.666	1.807	1.988	2.178	2.374	2.580
39.0	25°	0.719	0.846	0.981	1.118	1.265	1.419	1.451	1.578	1.746	1.928	2.110	2.293
45.0	30°	0.584	0.700	0.822	0.950	1.085	1.226	1.254	1.374	1.527	1.688	1.854	2.060
51.6	35°	0.460	0.567	0.682	0.798	0.924	1.052	1.079	1.190	1.332	1.478	1.632	1.796
58.6	40°	0.350	0.448	0.554	0.663	0.724	0.897	0.922	1.022	1.154	1.291	1.443	1.583

* = Inches of mercury below one standard atmosphere (29.92 in.).

The values of the factors for determining the volumetric efficiency due to clearance, F, for the assumed conditions of pressures and temperatures in the evaporator and condenser, are shown in Table 27 and Fig. 27. The actual efficiency of the compressor depends upon the efficiency due to superheating and that due to clearance. This may be expressed as follows:

$$E_a = E_s \times E_c$$

where E_a = actual volumetric efficiency
E_s = volumetric efficiency due to superheating
E_c = volumetric efficiency due to clearance

The mean effective pressure of ammonia in a compressor cylinder is also affected by the amount of clearance. The mean effective pressure with clearance is equivalent to the mean effective pressure without clearance multiplied by the volumetric efficiency due to clearance. This may be stated as follows:

$$m.e.p._c = m.e.p._o \times E_c$$

where $m.e.p._c$ = mean effective pressure with clearance, c
$m.e.p._o$ = mean effective pressure with zero clearance
E_c = volumetric efficiency due to clearance

The true volumetric efficiency of a compressor cylinder is determined by two factors principally. These are the superheating and clearance effects. The vapor entering during the suction stroke from the evaporator becomes superheated, due to the absorption of heat from the hot cylinder walls, piston and valves. The magnitude of the volumetric efficiency due to superheating seems to depend upon the ratio of compression, the type of compressor, and the relative speed of the compressor. The relation of these variables for ammonia as the refrigerant is shown by Fig. 28. The ratio of compression is obtained by dividing the absolute condenser pressure by the absolute suction pressure. Fig. 28 shows how the volumetric efficiency due to superheating, E_s, decreases as the ratio compression increases. The single-acting compressors, having a unindirectional flow of ammonia, possess efficiencies which are somewhat above those of the double-acting compressors, as shown by Fig. 28.

The volumetric efficiency due to superheating increases with the increase of the compressor speed. The relative efficiencies of the compressors of different speed classifications are shown by Fig. 28. The curves in Fig. 28 for the volumetric efficiencies due to superheating for the slow-speed vertical, single-acting and slow-speed horizontal, double-acting compressor were derived from tests performed by the York Manufacturing Company. The curves for high-speed horizontal, double-acting compressors were derived from manufacturers' ratings of commercial compressors. The curve for high-speed vertical, single-acting compressors was derived by mathematical analysis.

Actual Cylinder Displacement.—Since the weight of the vapor actually removed from the evaporator is less than the apparent, due to the fact that the volumetric efficiency is always less than 100 per cent, the actual piston displacement must be larger than the theoretical amount. To secure the estimate of the actual displacement, the theoretical displacement must be divided by the volumetric efficiency.

In case of the ammonia compressor, the theoretical displacement was 6750 cu. in. per min. per ton, at 0° F. suction and 86° F. condensing *(pg. 141)*. The actual displacement is obtained as follows, assuming a volumetric efficiency of 80 per cent:

$$\frac{6750}{0.80} = 8437 \text{ cu. in. per min. per ton}$$

Due to the fact that part of the displacement of the compressor is rendered ineffective on account of unavoidable losses in an actual compressor cylinder, the theoretical displacements must be increased in proportion to the volumetric efficiency of the cylinder. This may be expressed in symbols as follows:

$$A.D. = \frac{T.D.}{E_v} = \frac{200\,V}{(H_1 - h_2)E_v}$$

in which $A.D.$ = actual displacement, cu. ft. per ton of refrigeration per min.
E_v = volumetric efficiency.

In vertical single-acting compressors, in which the clearance has been reduced to the minimum workable amount, and in which the principal loss is due to superheating, the actual displacement is found as follows:

$$A.D. = \frac{T.D.}{E_s} = \frac{200 \times V}{(H_1 - h_2)\,E_s}$$

in which E_s = volumetric efficiency due to superheating.

For the standard conditions of 5° F. evaporating temperature and 86° F. condensing temperature, a slow-speed vertical single-acting ammonia compressor will have a volumetric efficiency of approximately 83.4 per cent. Hence, the actual displacement is found as follows:

$$A.D. = \frac{3.436}{0.834} = 4.120 \text{ cu. ft. per ton of refrigeration per min.}$$

Actual piston displacements of vertical single-acting compressor are shown by Table 28.

It is well to observe the relative proportions of ammonia compressor cylinders and the maximum allowable revolutions per minute.

TABLE 28.—ACTUAL DISPLACEMENTS OF SINGLE-ACTING AMMONIA COMPRESSORS CU. IN. PER MIN. PER TON OF REFRIGERATION.

Gauge Pressure Lbs. per sq. in.	Saturated Evaporating Temperature	Gauge Pressures, lbs. per sq. in. 92.9	103.1	114.1	125.8	138.3	151.7	154.5	165.9	181.1	197.2	214.2	232.3
		Saturated Condensing Temperatures 60°F.	65°	70°	75°	80°	85°	86°	90°	95°	100°	105°	110°
8.7″	−40°F.	23030	23380	23670	24500	25200	25780	25900	26450	27180	28160	28830	30220
5.4″	−35°	19850	20000	20440	20880	21380	21940	22100	22470	23250	23950	24590	25670
1.6″	−30°	16780	17030	17510	17930	18280	18800	18960	19280	19920	20480	21200	21930
1.3	−25°	14570	14740	15020	15380	15710	16150	16300	16510	17080	17560	18160	18780
3.6	−20°	12450	12710	12980	13320	13580	13910	14010	14350	14700	15160	15580	16240
6.2	−15°	10810	11030	11280	11520	11800	12080	12160	12430	12760	13140	13530	14000
9.0	−10°	9410	9610	9805	10030	10260	10510	10580	10820	11130	11430	11730	12330
12.2	− 5°	8220	8390	8570	8780	8950	9150	9245	9430	9670	9975	10270	10610
15.7	0°	7215	7350	7530	7588	7855	8050	8118	8250	8486	8710	8988	9290
19.6	5°	6350	6455	6610	6758	6910	7080	7126	7370	7470	7650	7881	8170
23.8	10°	5508	5712	5840	6032	6101	6248	6402	6412	6580	6880	7000	7212
28.4	15°	4961	5060	5196	5295	5411	5536	5572	5688	5832	5990	6198	6382
33.5	20°	4420	4508	4620	4708	4830	4930	4960	5062	5199	5349	5505	5679
39.0	25°	3942	4038	4130	4211	4318	4409	4430	4528	4658	4789	4920	5044
45.0	30°	3530	3608	3686	3757	3882	3956	3982	4058	4168	4279	4421	4565
51.6	35°	3166	3241	3311	3379	3483	3543	3568	3658	3756	3867	3964	4111
58 6	40°	2850	2924	2987	3048	3130	3222	3240	3309	3398	3478	3616	3720

″ = Inches of mercury below one standard atmosphere (29.92 in.).

Ammonia compressors of the slow-speed type generally have strokes of pistons which are 1.5 to 2.0 times the diameter of the cylinder, while compressors of the high-speed type will have strokes which are 1.0 to 1.5 times the diameter of the cylinder. The maximum allowable revolutions per min. for slow-speed compressors may be determined from Gardner's rule as follows:

$$r.p.m._{s.s.} = \frac{376}{\sqrt{s}}$$

r.p.m.s.s. = maximum speed of slow-speed compressors
where s = stroke of piston in inches
376 = a constant

Similarly, the maximum allowable revolutions per min. for high-speed compressors may be determined as follows:

$$r.p.m._{h.s.} = \frac{850}{\sqrt{s}}$$

where 850 = a constant

Compression of Vapor.—The compressor draws into its cylinder the vapor from the evaporator during the suction stroke, thereby removing the vapor from the evaporator and hence producing the desired refrigerating effect. At the end of the suction stroke the suction valve closes, trapping the vapor in the cylinder. The piston reverses its motion and the compression stroke is begun. The entrapped refrigerant is compressed until the pressure becomes slightly above that of the condenser; at this point, which is at about the ¾ point of the compression stroke, the discharge valve opens and the hot gas under high pressure is forced into the condenser. The gas becomes heated, since it requires the expenditure of mechanical energy or work to compress it from the low to the high pressure in the compressor cylinder.

In considering the nature of the action in the compressor cylinder, it is assumed that there is no transmission of heat between cylinder walls and the refrigerant, or, in other words, the compression takes place in a non-conducting envelope. When a vapor or gas is compressed or expanded without the loss or gain of heat, by radiation or conduction, the compression or expansion is said to be *adiabatic*.

It is further assumed that the refrigerant enters the compressor cylinder during the suction stroke in the form of a dry and saturated vapor; that is, it is in the form of a vapor with no suspended particles of liquid, and is at the temperature corresponding to the pressure on the refrigerant in the evaporator. Thus the refrigerant is not superheated. These considerations determine the condition of the refrigerant at the beginning of the compression stroke.

The gas, at the end of compression, is highly superheated, as previously indicated. The temperature will be many degrees above the saturation temperature due to the pressure, and the heat content will be increased in direct proportion to the amount of energy required to compress the vapor from the low to the high pressure.

In order to conveniently determine the condition of the vapor after compression, use may be made of the property "entropy." It will be remembered that entropy is a mathematical ratio obtained by dividing the amount of heat given to a substance at a given condition by the absolute temperature during the heat transfer. It is apparent, since there is no heat transfer during the adiabatic compression of a vapor, that the entropy will remain constant during the compression. Thus, the entropy of the vapor at beginning of the stroke and the entropy of the gas at the end of the compression are the same.

To illustrate the foregoing principles in a more material manner, ammonia will be assumed to be the working substance. Then, as soon as the pressure and temperature ranges are known, it is possible by reference to the tables of saturated and superheated ammonia to determine the exact state of the ammonia before and after compression. Thus, the condition of the ammonia evaporating at 5° F. and condensing at 86° F. may be considered. These temperatures correspond approximately to the absolute pressure of 34 and 169 lbs. per sq. in. respectively. The properties of saturated ammonia are indicated by Tables 6, 7 and 8 of Chapter III, while the properties of superheated ammonia are shown by Table 10, of Chapter III. In the Table 10, which contains the properties of the superheated vapor, "V," specific volume in cu. ft. per lb.; "H," heat content Btu. per lb.; "S," entropy. To use this table, the vertical section of the values corresponding to the absolute pressure is noted. Then, by reading down the table vertically until the corresponding value of the entropy "S," is found, the temperature, volume, and heat content corresponding to the pressure and entropy may be found. The condition before compression may be found in Tables 6, 7 and 8 of Chapter III.

The following tabulation will show the various properties of ammonia under the above conditions of temperature and pressure:

Condition Before Compression:	
Temperature of saturated vapor	5° F.
Absolute pressure of saturated vapor	34.27 lbs.
Approximate gauge pressure of saturated vapor	19.6 lbs.
Specific volume per pound	8.150 cu. ft.
Heat content of saturated vapor	613.3 Btu.
Entropy of saturated vapor	1.3253
Condition After Compression:	
Entropy of superheated vapor	1.3253
Absolute pressure of vapor	169 lbs.

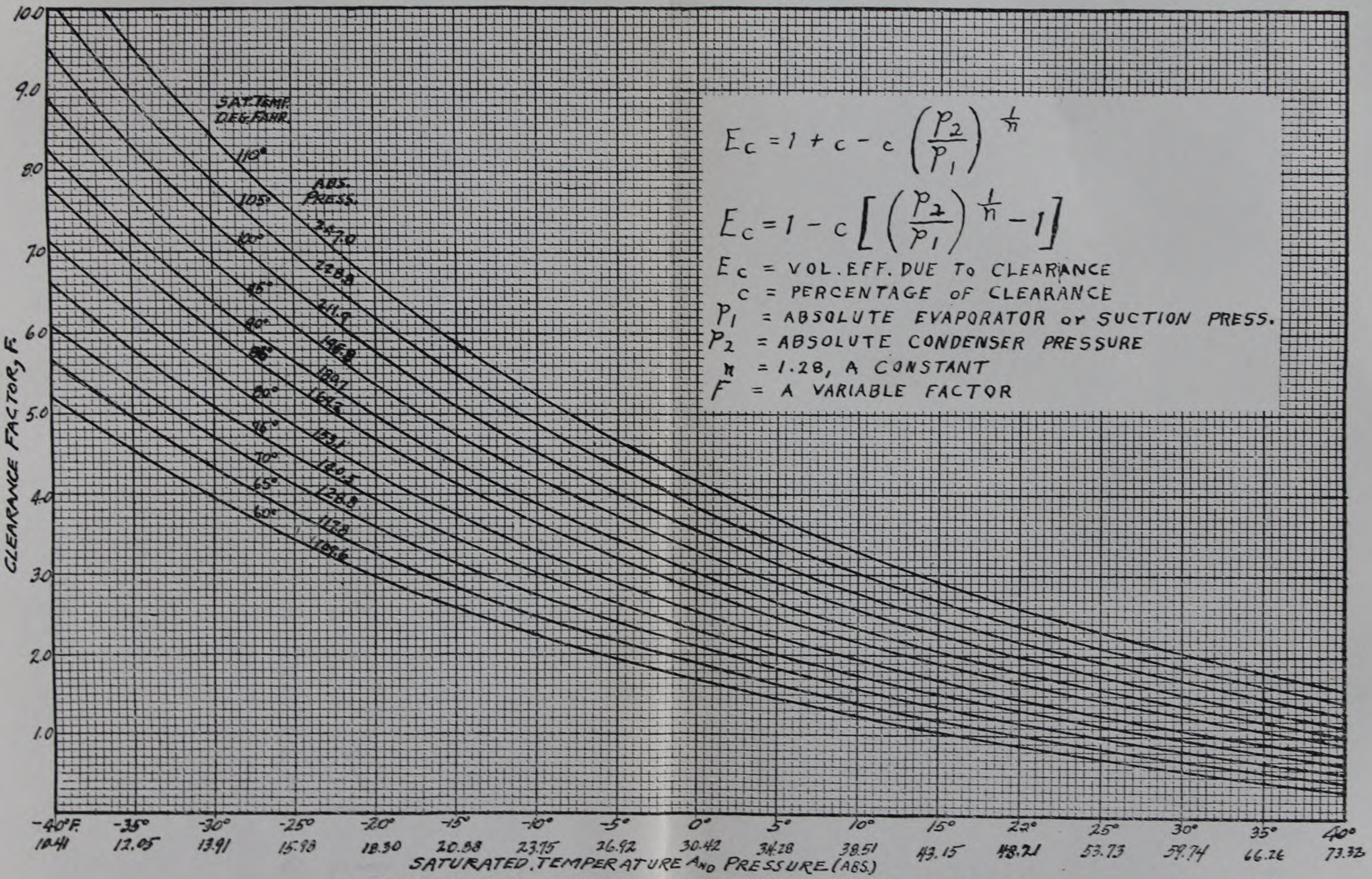


Fig. 27.—Factors for Calculating Volumetric Efficiencies Due to Clearances.

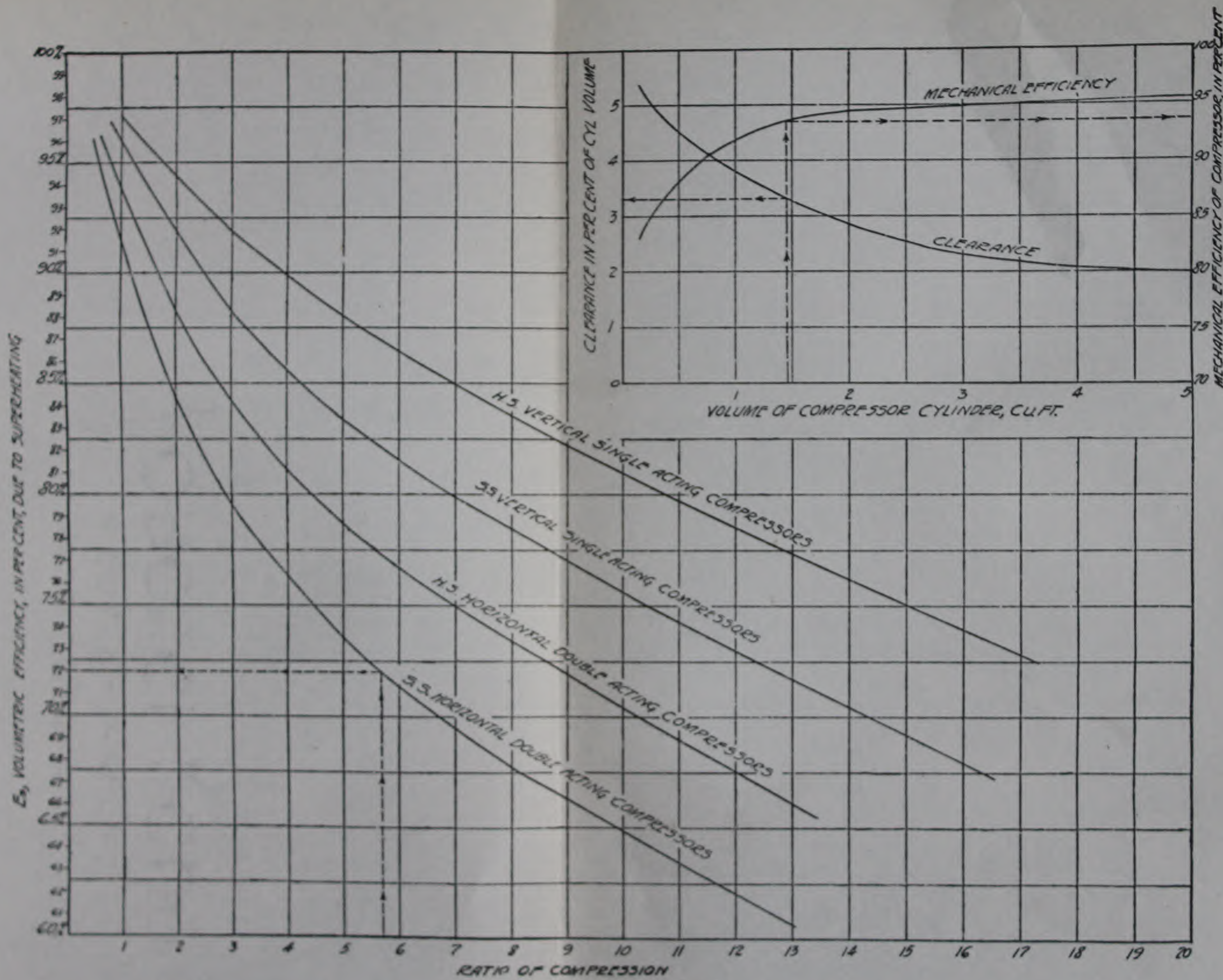


Fig. 28.—Compressor Efficiencies.

Approximate gauge pressure of vapor	155 lbs.
Temperature of saturated vapor	86° F.
Temperature of superheated vapor	210° F.
Degrees of superheat	124° F.
Specific volume per pound	2.346 cu. ft.
Heat content of superheated vapor	712.9 Btu.

Power Requirements.—It is evident that work must be expended to compress the refrigerant from the pressure of the evaporator to the pressure of the condenser. It is further evident that all of the work appears in the compressed gas in the form of heat, since the compression is adiabatic. The amount of power may be found by remembering that work and heat are inter-convertible in a constant ratio. This is the mechanical equivalent of heat, as discussed in Chapter II. Thus 778 ft-lbs. are equivalent to one Btu. Now, since one horsepower is the performance of work at the rate of 33,000 ft-lbs. per min., the heat equivalent of one horsepower is equal to $33{,}000 \div 778 = 42.42$ Btu. per min.

It is obvious that the difference of the heat contents before and after compression will give the heat equivalent of the work of compressing the refrigerant from the evaporator to the condenser pressure.

In the case of ammonia in the foregoing example, the heat content of the vapor before compression was 613.3 Btu. per lb. and the heat content of the gas after compression was 712.9 Btu. per lb.—giving a difference of $712.9 - 613.3 = 99.6$ Btu. per lb. The horsepower required to compress one pound of ammonia per minute will be found by dividing 99.6 by 42.42. Thus

$$99.6 \div 42.42 = 2.35 \text{ hp. per lb. of ammonia.}$$

Since the amount of ammonia to be evaporated has been found to be 0.422 lbs. per ton of refrigeration, the required horsepower per ton of refrigeration may be found as follows:

$$2.35 \times 0.422 = 0.989 \text{ hp. per ton of refrigeration.}$$

It must be noted that the above determination does not take into consideration the volumetric or mechanical efficiency of the compressor, and therefore the above result should be termed the theoretical indicated horsepower per ton of refrigeration.

Theoretical Power Requirements.—The theory and actual calculation of power requirements are simplified by the assumption that the comparison is adiabatic, that is, at constant entropy. Consequently, the heat equivalent of the work of compression is equal to the difference of heat contents of the refrigerant just before and after the compression. Actually, the compressions will approach closely the adiabatic process when cycles are performed rapidly and when the

vapor returns to the compressor in a nearly dry and saturated state. It is also assumed that there are no pressure changes in the system except in the compressor and expansion or throttle valve. The theoretical indicated horsepower may be derived, then, as follows:

$$t.i.hp. = (H_2 - H_1) \times \frac{200}{(H_1 - h_2)} \times \frac{777.46}{33000}$$

$$t.i.hp. = 4.712 \frac{(H_2 - H_1)}{(H_1 - h_2)}$$

$t.i.hp.$ = theoretical indicated horsepower per ton of refrigeration.
in which H_2 = heat content of refrigerant leaving compressor.

For ammonia at the standard conditions the horsepower is found as follows:

$$t.i.hp. = 4.712 \left(\frac{712.9 - 613.3}{613.3 - 138.9} \right) = 0.989$$

Theoretical indicated horsepowers per ton are given in Table 29.

Actual Indicated Power Requirements.—If the m.e.p. is determined, the power requirements may be readily found. The total force opposing the motion of the piston may be found by multiplying the area of the piston by the m.e.p. Thus:

$$\text{Force} = \text{m.e.p.} \times \text{area of piston}$$
$$\text{or } F = P \times A.$$

The work performed per working stroke may be had by multiplying the total force by the length of the stroke in feet. Thus:

$$\text{Work} = \text{force} \times \text{distance}$$
$$\text{or } W_1 = (P \times A) \times L.$$

The total work per minute may be obtained by multiplying the work per stroke by the total number of working strokes per minute. Thus, work per minute in ft-lbs. in the single-acting compressor is equal to

$$\text{Total work} = \text{work} \times \text{r.p.m.}$$
$$\text{or } W_2 = (P \times A) \times L \times N.$$

The horsepower may be obtained by dividing the total work in ft-lb. per min. by the number of ft-lbs. in one horsepower, which is 33,000. Thus,

$$\text{Horsepower} = \frac{\text{m.e.p.} \times \text{area} \times \text{length} \times \text{r.p.m.}}{33{,}000}$$

$$\text{or hp.} = \frac{P \times A \times L \times N}{33000}$$

Where hp. = horsepower
P = m.e.p. in lbs. per sq. in.
A = area of piston in sq. in.
L = length of stroke in ft.
N = r.p.m. = revolutions per min.

The actual indicated horsepower requirements are obtained by dividing the theoretical amount by the volumetric efficiency, thus:

$$a.i.hp. = \frac{a.i.hp.}{E_v} = \frac{4.712}{E_v}\left(\frac{H_2 - H_1}{H_1 - h_2}\right)$$

in which $a.i.hp.$ = actual indicated horsepower per ton of refrigeration.

For vertical single-acting compressor with practically no clearance, the actual indicated horsepower is given as follows:

$$a.i.hp. = \frac{t.i.hp.}{E_s} = \frac{4.712}{E_s}\left(\frac{H_2 - H_1}{H_1 - h_2}\right)$$

For standard conditions, and for ammonia, it is found as follows:

$$a.i.hp. = \frac{0.989}{0.834} = \frac{4.712}{0.834}\left(\frac{712.9 - 613.3}{613.3 - 138.9}\right) = 1.186$$

Input Horsepower.—The input power is that power taken from the electric power lines in case of electric drive, or the indicated power of the steam engine in case of steam engine drive. The input exceeds the actual indicated horsepower requirements by the amount due to losses of friction, windage, etc., which usually amounts to 20 per cent in medium sized plants. This corresponds to a combined mechanical efficiency of 83.3 per cent.

The input horsepower requirements per ton of refrigeration are found as follows:

$$Input\ hp. = \frac{a.i.hp.}{E_m} = \frac{4.712}{E_s \times E_m}\left(\frac{H_2 - H_1}{H_1 - h_2}\right)$$

$$Input\ hp. = \frac{a.i.hp.}{0.833} = \frac{5.656}{E_s}\left(\frac{H_2 - H_1}{H_1 - h_2}\right)$$

in which E_m = mechanical efficiency of compressor and prime mover.
$Input\ hp.$ = input horsepower per ton of refrigeration.

For ammonia at standard conditions, the input power per ton refrigeration is found as follows:

$$Input\ hp. = \frac{1.186}{0.833} = \frac{5.656}{0.833}\left(\frac{712.9 - 613.3}{613.3 - 138.9}\right) = 1.424$$

TABLE 29.—THEORETICAL INDICATED HORSEPOWER PER TON OF REFRIGERATION BASED ON BUREAU OF STANDARDS AMMONIA TABLES.

Gauge Pressure Lbs. per sq. in.	Saturated Evaporating Temperature	Gauge Pressures, lbs. per sq. in. 92.9	103.1	114.1	125.8	138.3	151.7	154.5	165.9	181.1	197.2	214.2	232.3
		Saturated Condensing Temperatures 60°F.	65°	70°	75°	80°	85°	86°	90°	95°	100°	105°	110°
58.6	40°F.	0.2016	0.2410	0.3076	0.3620	0.4124	0.4706	0.4900	0.5312	0.5914	0.6520	0.7220	0.7860
51.6	35°	0.2542	0.3188	0.3692	0.4230	0.4810	0.5360	4.5530	0.5960	0.6610	0.7120	0.7910	0.8540
45.0	30°	0.3040	0.3590	0.4185	0.4790	0.5360	0.5960	0.6115	0.6580	0.7230	0.7860	0.8530	0.9210
39.0	25°	0.3666	0.4273	0.4823	0.5440	0.6080	0.6670	0.6825	0.7318	0.7954	0.8650	0.9430	1.0080
33.5	20°	0.4290	0.4890	0.5550	0.6190	0.6810	0.7420	0.7650	0.8170	0.8830	0.9510	1.0230	1.0920
28.4	15°	0.4988	0.5598	0.6242	0.6910	0.7563	0.8260	0.8386	0.8935	0.9630	1.0390	1.1150	1.1920
23.8	10°	0.5620	0.6242	0.6942	0.7629	0.8310	0.8913	0.9140	0.9660	1.0380	1.1140	1.1960	1.2740
19.6	5°	0.6140	0.6870	0.7545	0.8278	0.8900	0.9582	0.9835	1.0320	1.1090	1.1840	1.2700	1.3380
15.7	0°	0.6910	0.7610	0.8299	0.9030	0.9750	1.0380	1.0730	1.1220	1.1980	1.2820	1.3630	1.4370
12.2	− 5°	0.7820	0.8490	0.9230	0.9990	1.0720	1.1430	1.1610	1.2220	1.3060	1.3870	1.4780	1.5560
9.0	−10°	0.8750	0.9240	1.0030	1.0830	1.1540	1.2260	1.2500	1.3140	1.3930	1.4790	1.5680	1.6560
6.2	−15°	0.9540	1.0250	1.1090	1.1850	1.2640	1.3430	1.3650	1.4310	1.5150	1.6030	1.6960	1.7920
3.6	−20°	1.3700	1.1220	1.2020	1.2790	1.3630	1.4420	1.4700	1.5330	1.6200	1.7120	1.8160	1.9070
1.3	−25°	1.1340	1.2120	1.2960	1.3630	1.4660	1.5520	1.5800	1.6480	1.7410	1.8450	1.9310	2.0320
1.6″	−30°	1.2240	1.3050	1.3950	1.4910	1.5660	1.6550	1.6800	1.7540	1.8460	1.9450	2.0500	2.1530
5.4″	−35°	1.3300	1.4130	1.5050	1.5960	1.6830	1.7750	1.8010	1.8780	1.9780	2.0780	2.1830	2.2970
8.7″	−40°	1.4200	1.5050	1.6060	1.7030	1.7930	1.8900	1.9160	1.9940	2.0980	2.2000	2.3180	2.4260

″ = Inches of mercury below one standard atmosphere (29.92 in.).

The corresponding input kilowatts are obtained by multiplying the input horsepower by the constant 0.746:

$$Input\ kw. = \text{Input hp.} \times 0.746.$$

The corresponding horsepower-hours and kilowatt-hours per ton of refrigeration are found by multiplying the respective horsepowers and kilowatts by 24:

$$Kw\text{-}hr. = \text{Input kw.} \times 24.$$
$$Hp\text{-}hr. = \text{Input hp.} \times 24.$$

The horsepower required for a given compressor cylinder may also be found, for a single-acting cylinder, as follows:

$$\text{hp.} = \frac{P \times L \times A \times N}{396000}$$

in which
P = mean effective pressure in lbs. per sq. in.
L = length of stroke in inches.
A = area of piston in sq. in.
N = r.p.m.

The mean effective pressure, P, may be found as follows:

$$P = \frac{5.399\ (H_2 - H_1)}{V}$$

It should be noted that the foregoing new formula for the mean effective pressure is much simpler to use than the older formula which required the use of logarithms for its solution. It should be further noted, that the above new formula gives values for the mean effective pressure, which are accurate and thermodynamically consistent for all conditions of operation.

For standard conditions of 5° F. and 86° F. in evaporator and condenser, the mean effective pressure is found to be:

$$P = \frac{5.399\ (712.9 - 613.3)}{8.150} = 65.9 \text{ lbs.}$$

Table 30 shows the variation of the mean effective pressures for the different conditions.

By an inspection of Table 30 it will be noted that when the suction pressure is constant, an increase of condenser pressure increases slightly the horsepower required per ton, and that when the condenser pressure is constant, increasing the suction pressure decreases the horsepower requirements greatly.

As previously indicated, the theoretical indicated horsepower may be determined by means of the mean effective pressure of the am-

TABLE 30.—MEAN EFFECTIVE PRESSURES, LBS. PER SQ. IN. BASED ON BUREAU OF STANDARDS AMMONIA TABLES.

Gauge Pressure Lbs. per sq. in.	Saturated Evaporating Temperature	Gauge Pressures, lbs. per sq. in.											
		92.9	103.1	114.1	125.8	138.3	151.7	154.5	165.9	181.1	197.2	214.2	232.3
		Saturated Condensing Temperatures											
		60°F.	65°	70°	75°	80°	85°	86°	90°	95°	100°	105°	110°
58.6	40°F.	29.91	35.35	44.60	51.94	58.45	66.00	68.30	73.54	80.90	87.10	96.10	103 4
51.6	35°	34.22	42.35	48.54	54.96	61.76	68.05	69.70	74.70	81.90	87.10	95.50	101.8
45.0	30°	36.93	43.10	49.70	56.30	62.25	68.40	69.95	74.66	80.90	86.97	93.10	99.25
39.0	25°	40.20	46.37	51.74	57.70	63.70	68.97	70.50	74.90	80.30	86.30	92.87	98.00
33.5	20°	42.32	47.70	53.45	58.92	64.14	69.20	70.80	74.85	80.30	85.20	90.55	95.40
28.4	15°	43.19	49.03	54.15	59.17	64.00	69.00	70.34	73.80	78.60	83.70	88.55	93.50
23.8	10°	44.60	49.03	53.80	58.50	62.90	66.80	68.20	71.40	75.80	80.40	85.10	89.40
19.6	5°	44.00	48.18	52.30	56.65	60.24	64.10	65.60	68.15	72.30	76.25	80.70	84.10
15.7	0°	43.85	47.53	51.27	55.12	58.80	61.92	63.80	66.10	69.65	73.60	77.10	80 30
12.2	− 5°	43.80	47.05	50.65	54.20	57.40	60.45	61.30	63.90	67.40	70.70	74.30	77.20
9.0	−10°	42.60	45.50	48.77	52.00	54.80	57.52	58.45	60.80	63.75	66.83	69 90	72.84
6.2	−15°	41.90	44.56	47.62	50.27	53.10	55.60	56.44	58.51	61.22	63.92	66.80	69.60
3.6	−20°	40.10	42.76	45.27	47.80	50.30	52.60	53.30	55.20	57.70	60.05	62.85	65.15
1.3	−25°	38.50	40.65	42.94	44.69	47.56	49.60	50.25	52.10	54.25	56.72	58.63	60.90
1.6″	−30°	36.34	38.31	40.51	42.74	44.40	46.30	46.86	48.47	50.40	52.40	54.50	56 45
5.4″	−35°	34.40	36.21	38.10	39.87	41.58	43.30	48.83	45.30	47.05	48.78	50.24	52 42
8.7″	−40°	31.94	33.50	35.28	36.95	38.45	40.07	40.50	41.62	43.34	44.85	46.60	48 13

″ = Inches of mercury below one standard atmosphere (29.92 in.).

monia, or it may be determined by means of the variation of the heat contents.

The actual amount of displacement required in the compressor cylinder will be somewhat larger than the theoretical amount, since the volumetric efficiency is always less than 100 per cent. In order to secure the estimate of the actual amount of displacement, the theoretical displacements must be divided by the volumetric efficiency, which means that the theoretical amount is increased to make up for the loss of effective displacement due to the inherent defects of the compressor.

In similar manner, the brake horsepower required must be larger than the actual indicated horsepower, due to the fact that power is lost in overcoming the friction of the machine, etc. This loss of power in operating the machine is commonly termed "the mechanical efficiency of the machine." Thus:

$$b.hp. = \frac{\text{a.i.hp.}}{\text{m.e.c.}}$$

where *b.hp.* = brake horsepower
m.e.c. = mechanical efficiency of the compressor

After the brake horsepower of the compressor has been determined, it is a comparatively easy matter to determine the horsepower of the motor, or the horsepower of the engine required to drive the compressor. In the case of the motor drive, it is evident that the brake horsepower must be increased in proportion to the mechanical efficiency of the method of driving in order to determine the output horsepower required from the motor. In the case of steam-engine drive, the indicated horsepower of the steam-engine may be obtained by simply increasing the brake horsepower of the compressor in direct proportion to the mechanical efficiency of the steam-engine.

The brake horsepower of the compressor per ton of refrigeration is found by the formula:

$$b.hp. = \frac{a.i.hp.}{E_m} = \frac{4.712}{E_s \times E_m}\left(\frac{H_2 - H_1}{H_1 - h_2}\right)$$

in which E_m = mechanical efficiency of compressor.

Condensation of Vapor.—The gas leaving the compressor is forced into the condenser under high pressure and at a high temperature. The pressure is at such a point so that the mean temperature of the ammonia is a few degrees above the average temperature of the condenser water. The heat, therefore, flows from the refrigerant into the available condenser water, thereby cooling and condensing the refrigerant. In the first part of the condenser the hot discharge gases from

the compressor are cooled down to the saturation temperature. By further extraction of heat at a constant temperature, the medium will condense into its liquid form. Then by additional abstractions the liquid may be cooled a few degrees below the saturation temperature.

The case of ammonia working between 34.27 and 169.2 lbs. abs. will be considered, assuming that the liquid ammonia is after-cooled to 76°. The following tabulations show the condition of the ammonia at various points in the condenser:

Condition at Entrance to Condenser:	
Temperature	210° F.
Heat content of superheated vapor	712.9 Btu.
Condition at Entrance to Saturated Portion:	
Temperature	86° F.
Heat content, saturated vapor	631.5 Btu.
Condition at End of Saturated Portion:	
Temperature	86° F.
Heat content of liquid	138.9 Btu.
Condition at End of Condenser:	
Temperature	76° F.
Heat content of liquid	127.4 Btu.

The heat removed in the superheated portion is equal to 712.9 − 631.5 = 81.4; in the saturated portion, 631.5 − 138.9 = 492.6; in the aftercooling portion, 138.9 − 127.4 = 11.5; or the total cooling re- quired per pound of ammonia is equal to 81.4 + 492.6 + 11.5 = 585.5 Btu. The same result may be obtained by noting the difference of the heat content of the ammonia before and after the condenser; thus, 712.9 − 127.4 = 585.5 Btu., the necessary cooling effect required.

It is obvious that the condenser removes the heat absorbed in the evaporator and the heat added in the compressor. In the case of no aftercooling effect of the liquid, the total heat removed by the con- denser water is equal to 712.9 − 138.9 = 574.0 Btu. As previously determined, the heat removed by the ammonia from the refrigerator was 474.4 Btu. and the work of compression was equivalent to 99.6 Btu. Their sum is equal to 474.4 + 99.6 = 574.0 Btu. which corre- sponds to the above value.

Heat Removed in Condenser.—The heat removed in the condenser may be taken to be the heat equivalent of the actual work of compres- sor per min. per ton plus the heat absorbed in the evaporator per min. per ton of refrigeration. This may be stated in symbols as follows:

$$H_c = a.i.hp. \times 42.44 + 200$$

$$H_c = \frac{4.712}{E_s}\left(\frac{H_2 - H_1}{H_1 - h_2}\right) \times 42.44 + 200$$

$$\therefore H_c = 200\left[\left(\frac{H_2 - H_1}{H_1 - h_2}\right)\frac{1}{E_s} + 1\right]$$

For ammonia at the standard conditions of 5° and 86° F. in the evaporator and condenser respectively, and for vertical single-acting compressors, the heat to be removed in the condenser may be found as follows:

$$H_c = 200\left[\left(\frac{712.9-613.3}{613.3-138.9}\right)\frac{1}{0.834}+1\right]$$
$$=250.3 \text{ Btu. per ton of refrigeration per min.}$$

If the liquid ammonia is aftercooled to 75° F., the heat to be removed in the condenser is found as follows:

$$H_c = 200\left[\left(\frac{712.9-613.3}{613.3-126.2}\right)\frac{1}{0.834}+1\right]$$
$$=249.0 \text{ Btu. per ton of refrigeration per min.}$$

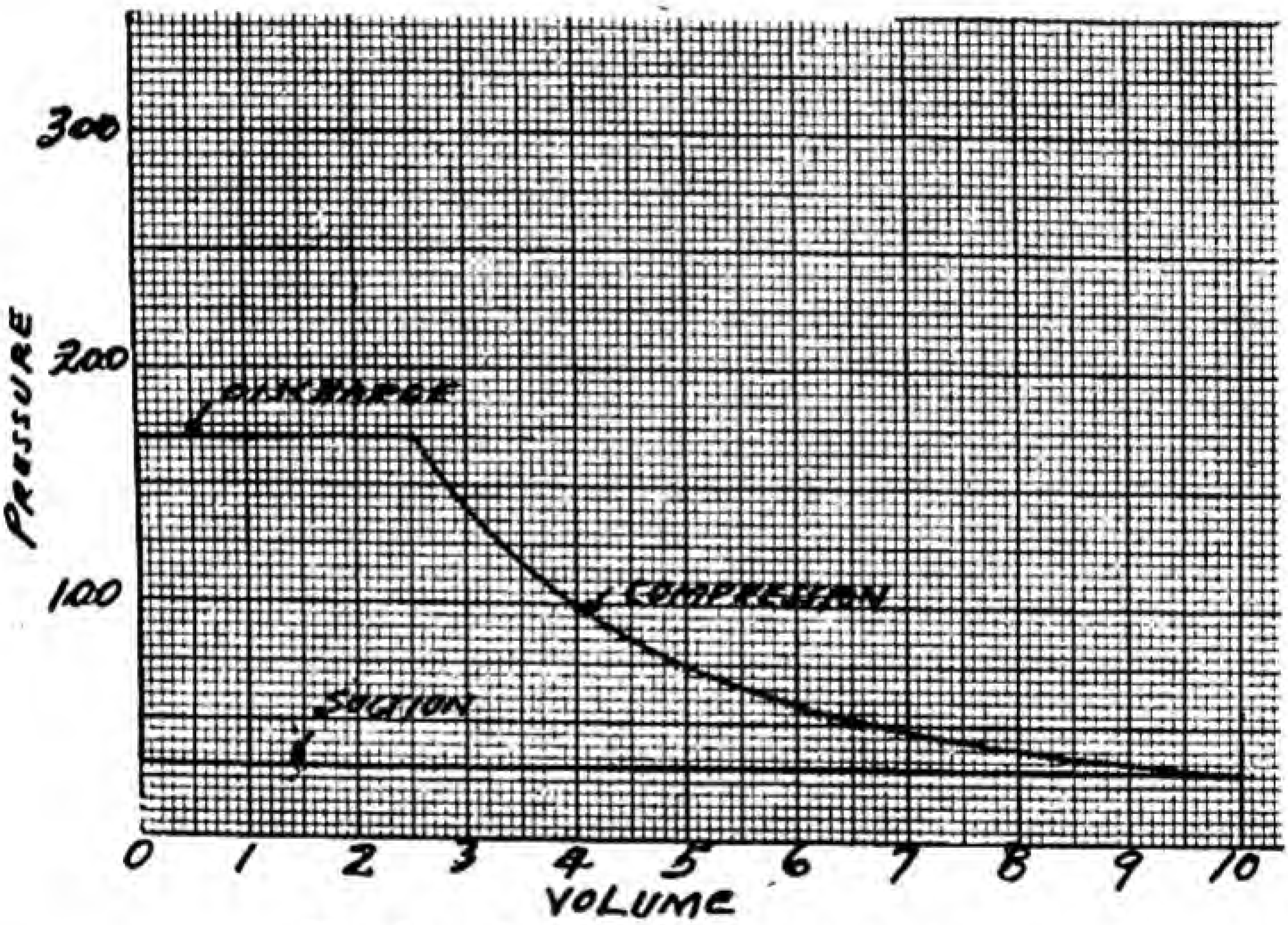


Fig. 29.—Pressure Volume Diagram.

Cooling Water Requirements.—From the foregoing it is apparent that the amount of heat to be removed from the refrigerant may be found by noting the difference between the heat content of the superheated gas as it enters the condenser and the heat content of the liquid leaving the condenser. Thus, in the previous example it was found that when the liquid is aftercooled to 76° F., the heat to be extracted from the ammonia was 712.9 − 127.4 = 585.5 Btu. per lb. It is, therefore, necessary to put enough water through the condenser to absorb this heat, heating the water a few degrees. Should the temperature of the water raise 8° F. in passing through the condenser, that is, heating

from 70° to 78° F., for example, the amount of water to be circulated or supplied will be found as follows, since the specific heat of water is 1:0:

$$\text{Heat removed by water} = (78° - 70°) \times 1 = 8 \text{ Btu. per lb.}$$

$$\text{Lbs. of water required} = \frac{585.5}{8} = 73.2 \text{ lbs. per lb. of ammonia.}$$

$$\text{Gallons of water} = \frac{73.2}{8.33} = 8.78 \text{ gallons per lb. of ammonia.}$$

$$\text{Gallons per ton of refrigeration} = 0.422 \times 8.78 = 3.70 \text{ g.p.m.}$$

Water Requirements for Condenser.—The water requirements for the condenser will depend upon the heat to be removed in the condenser and the relative temperature range of the water in passing through the condenser. The relationship may be stated as follows:

$$g.p.m. = \frac{H_c}{8.33 \times (t_1 - t_2)}$$

$$g.p.m. = \frac{200\left[\left(\dfrac{H_2 - H_1}{H_1 - h_2}\right)\dfrac{1}{E_s} + 1\right]}{8.33\,(t_1 - t_2)}$$

$$g.p.m. = \frac{24\left[\left(\dfrac{H_2 - H_1}{H_1 - h_2}\right)\dfrac{1}{E_s} + 1\right]}{(t_1 - t_2)}$$

in which $g.p.m.$ = gallons of water required per min. per ton of refrigeration.
t_1 = outlet temp. of water
t_2 = inlet temp. of water

For standard conditions of 5° and 86° F. saturated ammonia temperatures, water in at 70° F. and out at 80° F., with the liquid ammonia aftercooled to 75° F., the water requirements are found as follows:

$$g.p.m. = \frac{24\left[\left(\dfrac{712.9 - 613.3}{613.3 - 126.2}\right)\dfrac{1}{0.834} + 1\right]}{(80 - 70)}$$

$= 3.005$ per ton refrigeration.

Heat Removed in Different Parts of Condenser.—The quantities of heat to be removed in the superheater, liquefier, and aftercooler portions of a condenser of the counter-current type are given by the following equations:

For liquefier: $$H_L = \frac{200\,L}{(H_1 - h_2)}$$

in which L = latent heat of condensation at the condenser pressure.

For aftercooler:

$$H_a = \frac{200\,(h_2 - h_1)}{(H_1 - h_2)}$$

in which h_2 = heat content of refrigerant sat. cond. temp.

For superheater: $H_s = H_c - (H_L + H_a)$

The heats, H_L, H_a and H_s are the amounts of heat in Btu. to be removed per min. per ton of refrigeration.

Surface Requirements for Condenser.—The heat transmitting surface required for the condenser will depend upon the amount of heat to be removed, the mean temperature difference between the condensing refrigerant and the water, and the heat transfer rate or coefficient of the effective surface.

The average mean temperature difference for the whole condenser may be calculated by the following formula:

$$t_d = \frac{H_c}{\dfrac{H_a}{t_a} + \dfrac{H_L}{t_L} + \dfrac{H_s}{t_s}}$$

in which t_d = true mean temp. difference for whole condenser.
t_a = mean temp. diff. of aftercooler.
t_L = mean temp. diff. of liquefier.
t_s = mean temp. diff. of superheater.

The mean temperature differences of the aftercooler, liquefier, and superheater portions, t_a, t_L, and t_s, may be calculated as follows, for a counter-current condenser such as a double-pipe condenser:

$$t_a = 0.4342\frac{(t_4 - t_6) - (t_5 - t_1)}{\log.\dfrac{(t_4 - t_6)}{(t_5 - t_1)}}$$

$$t_L = 0.4342\frac{(t_4 - t_6) - (t_4 - t_7)}{\log.\dfrac{(t_4 - t_6)}{(t_4 - t_7)}}$$

$$t_s = 0.4342\frac{(t_3 - t_2) - (t_4 - t_7)}{\log.\dfrac{(t_3 - t_2)}{(t_4 - t_7)}}$$

in which t_1 = temp. of water at inlet to condenser.
t_2 = temp. of water at outlet of condenser.
t_3 = temp. of refrigerant from compressor for adiabatic compression.
t_4 = temp. of condensation.
t_5 = temp. of aftercooled refrigerant.
t_6 = temp. of water at outlet of aftercooler.
t_7 = temp. of water at outlet of liquefier.

In the foregoing, the temperatures of the water at the end of the aftercooler and superheater portions are found as follows:

$$t_6 = t_1 + \frac{H_a}{H_c}(t_2 - t_1)$$

$$t_7 = t_1 + \left(\frac{H_a + H_L}{H_c}\right)(t_2 - t_1)$$

The surface requirements may then be calculated as follows:

$$A = \frac{H_c}{Kt_d} = \frac{200\left[\left(\frac{H_2 - H_1}{H_1 - h_2}\right)\frac{1}{E_s} + 1\right]}{Kt_d}$$

in which A = area of surface in sq. ft. per ton of refrigeration.
K = heat transfer coefficient, Btu per hr. per sq. ft. per deg. temp. diff.

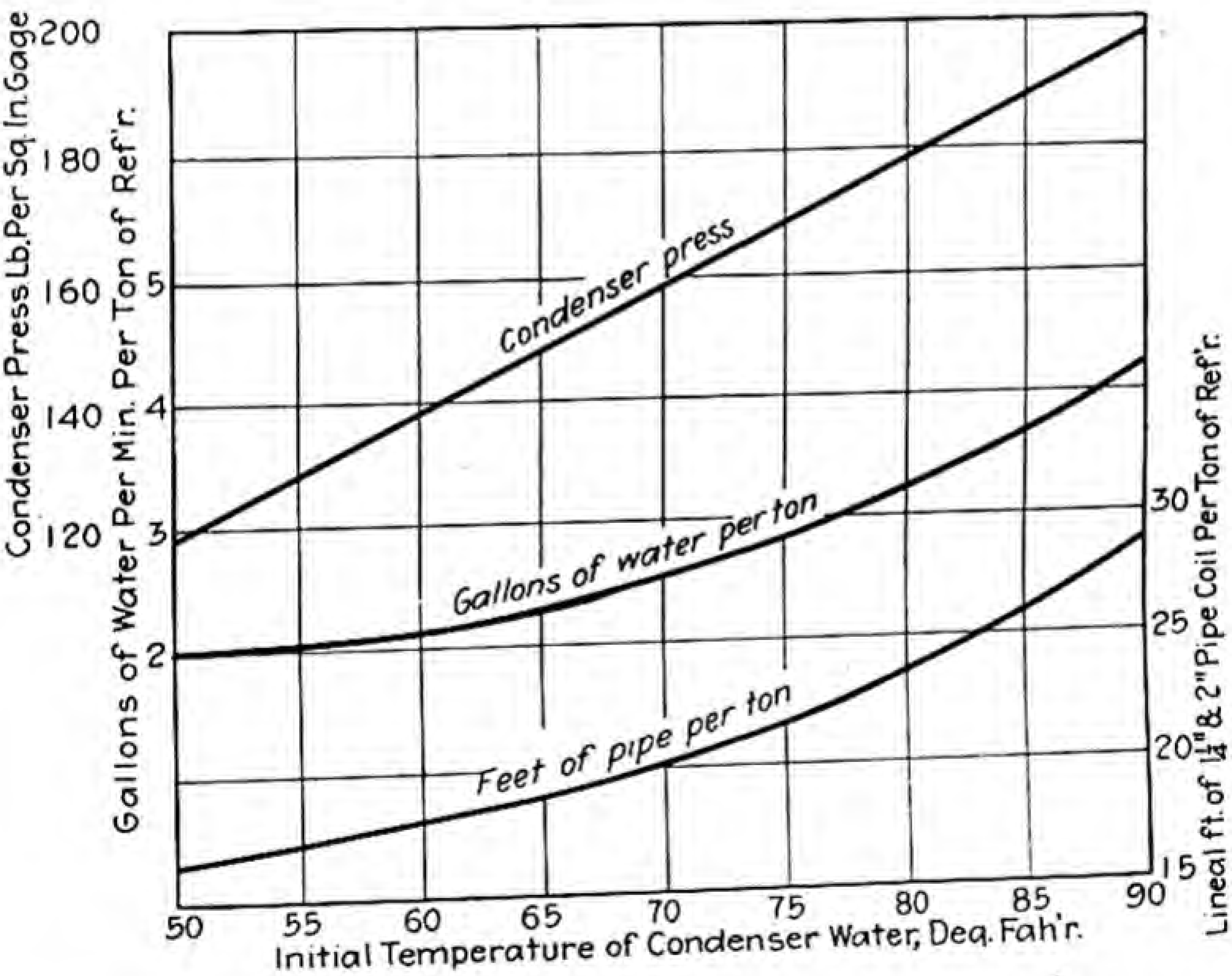


Fig. 30.—Operating Characteristics of Double-Pipe Condenser.

The heat transfer coefficients will depend upon the disposition of the condenser surface, the velocity of the fluids, surface effect, etc. The following values are some conservative estimates of the coefficients in the different forms of condensers:

Kind of Condenser	Heat transfer coefficient, K, Btu. per sq. ft. per hr. per deg. of temp. diff.
Double-pipe	150
Horizontal Shell-and-Tube	150
Vertical Shell-and-Tube	125
Atmospheric	50
Submerged	30

Condenser Calculation.—It is desired to determine the surface requirements for a double-pipe condenser, which is supplied with water at 70° F., heats the water to 80° F., and operates on a system having a 5° F. evaporating temperature and 86° F. condenser temperature. The various quantities are calculated as follows, when the liquid is aftercooled to 75° F.:

$$H_c = 249$$

$$H_a = (138.9 - 126.2)\frac{200}{613.3-126.2} = 5.22$$

$$H_L = \frac{200 \times 492.6}{613.3-126.2} = 202.2$$

$$H_s = 249-(202.2+5.22) = 41.58$$

$$t_1 = 80^\circ$$

$$t_2 = 70^\circ$$

$$t_3 = 210^\circ \text{ from ammonia table}$$

$$t_4 = 86^\circ$$

$$t_5 = 75^\circ$$

$$t_6 = 70^\circ + \frac{5.22}{249}(80-70) = 70.21^\circ$$

$$t_7 = 70^\circ + \frac{(5.22+202.2)\times 10}{249} = 78.33^\circ$$

$$t_a = 0.4342\frac{(86-70.21)-(75-70)}{\log.\dfrac{(86-70.21)}{(75-70)}} = 9.36^\circ$$

$$t_L = 0.4342\frac{(86-70.21)-(86-78.33)}{\log.\dfrac{(86-70.21)}{(86-78.33)}} = 11.25^\circ$$

$$t_s = 0.4342\frac{(210-80)-(86-78.33)}{\log.\dfrac{(210-80)}{(86-78.33)}} = 43.2^\circ$$

$$t_d = \frac{249}{\dfrac{5.22}{9.36}+\dfrac{202.2}{11.25}+\dfrac{41.58}{43.2}} = 12.77^\circ$$

$$A = \frac{249\times 60}{150\times 12.77} = 7.80 \text{ sq. ft.}$$

Condenser Pressures, Water Temperatures, Etc.—Figs. 30, 31, 32, 33, and 34 have been prepared to show graphically how the amounts of surface, condenser pressures, and quantities of water for double-pipe, standard atmospheric, bleeder-type, single-pass shell-and-tube and 7-pass multi-tube type condensers vary with the increase of the initial temperature of the condenser water. Generally speaking, under average conditions these amounts of surfaces and water are to be recommended for plants in the United States in order to obtain maximum economy. The charts give the surface in lineal feet of condenser pipe.

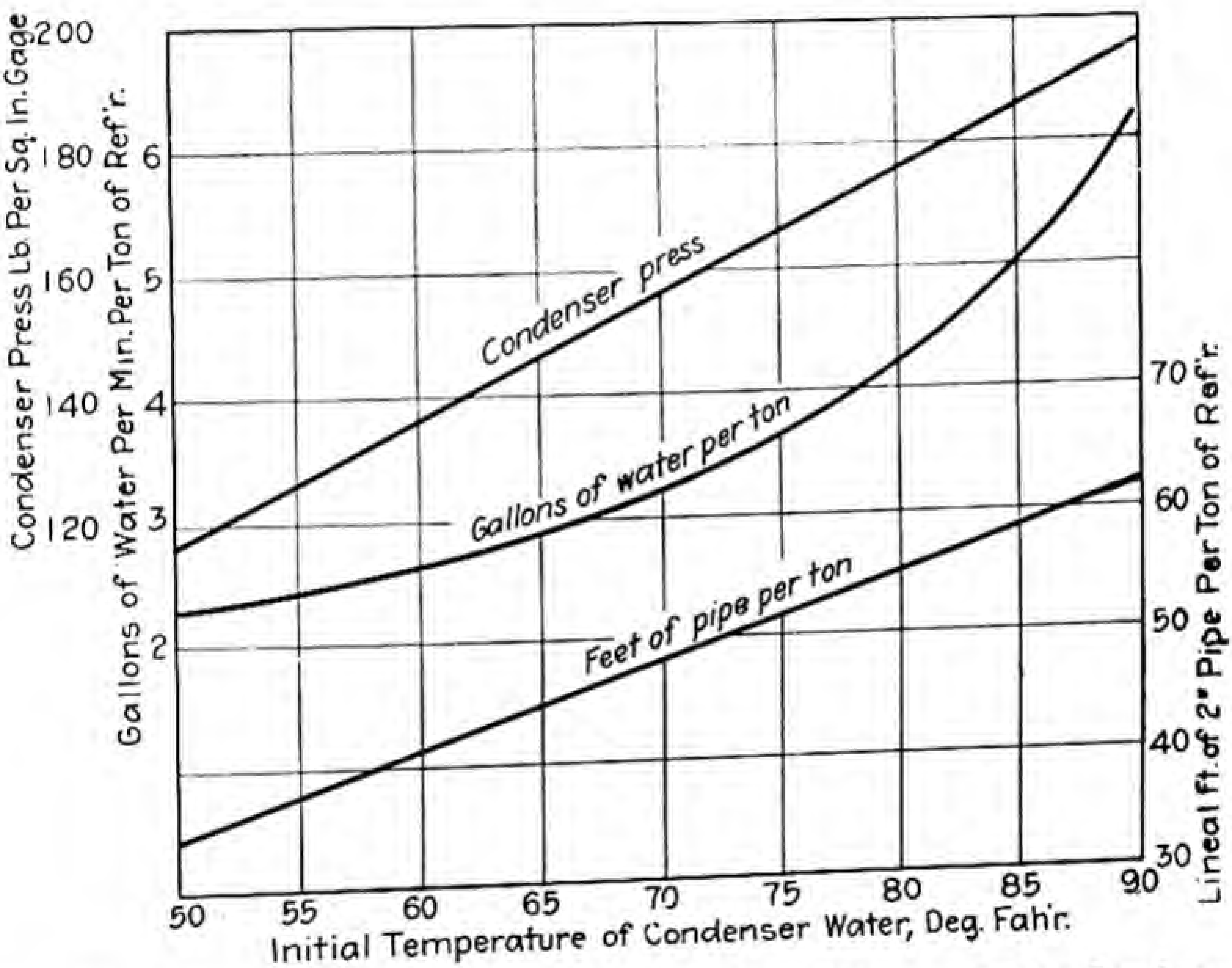


Fig. 31.—Operating Characteristics of Standard Atmospheric Condenser.

Figs. 30, 31, 32, 33 and 34 show the amount of water that should be pumped to the condenser under average conditions. Those figures also show how the quantity of water should vary with the increase of initial temperature of the water.

These figures indicate the approximate condenser pressure that may be expected with the corresponding amounts of surface and water for the various types of common condensers. The condenser pressure in an actual plant may vary somewhat from those indicated in the figures, which may be due to the general operating condition, such as the presence of air in the system, amount of oil in the condenser, etc.

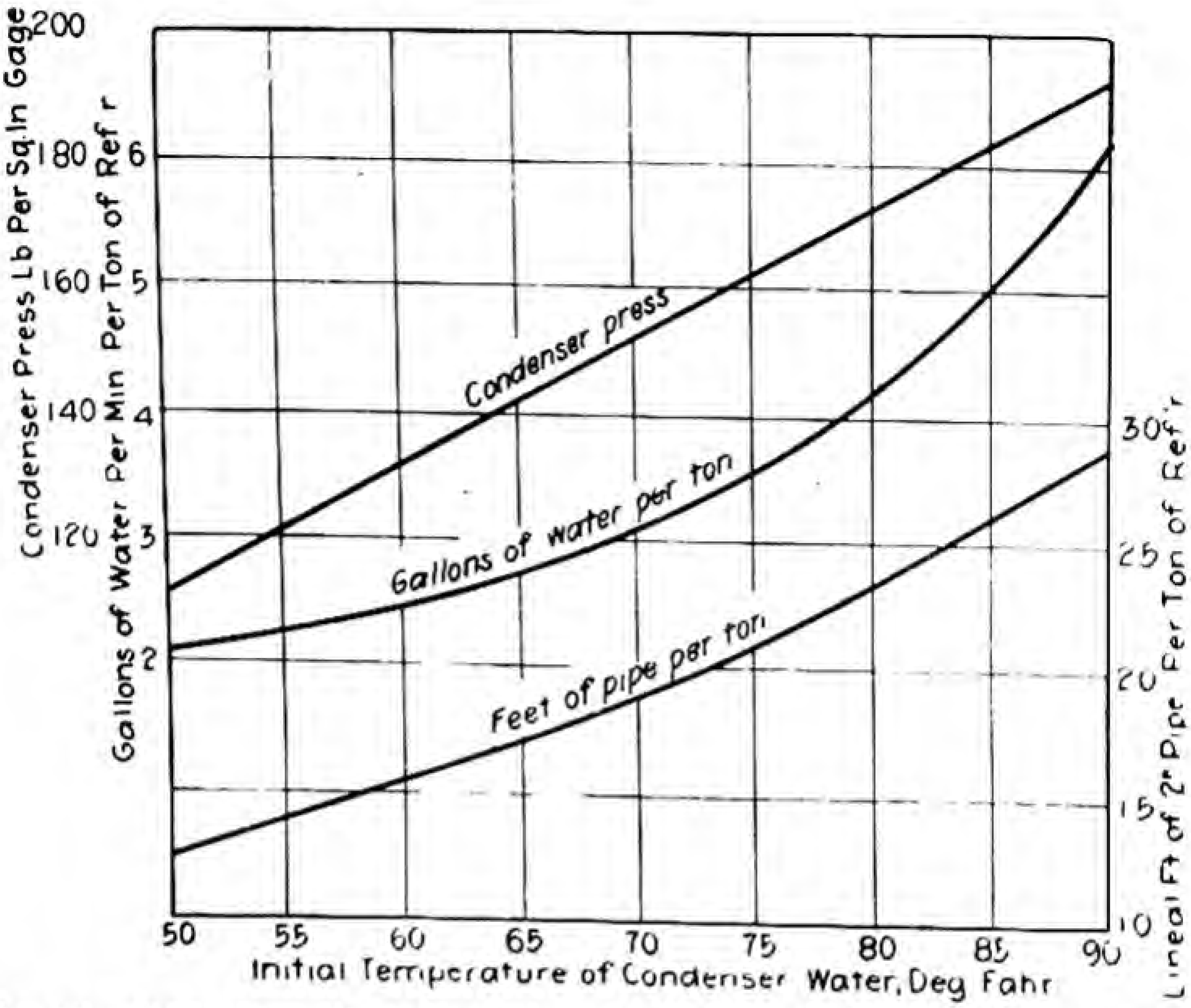


Fig. 32.—Operating Characteristics of Bleeder Type Ammonia Condenser.

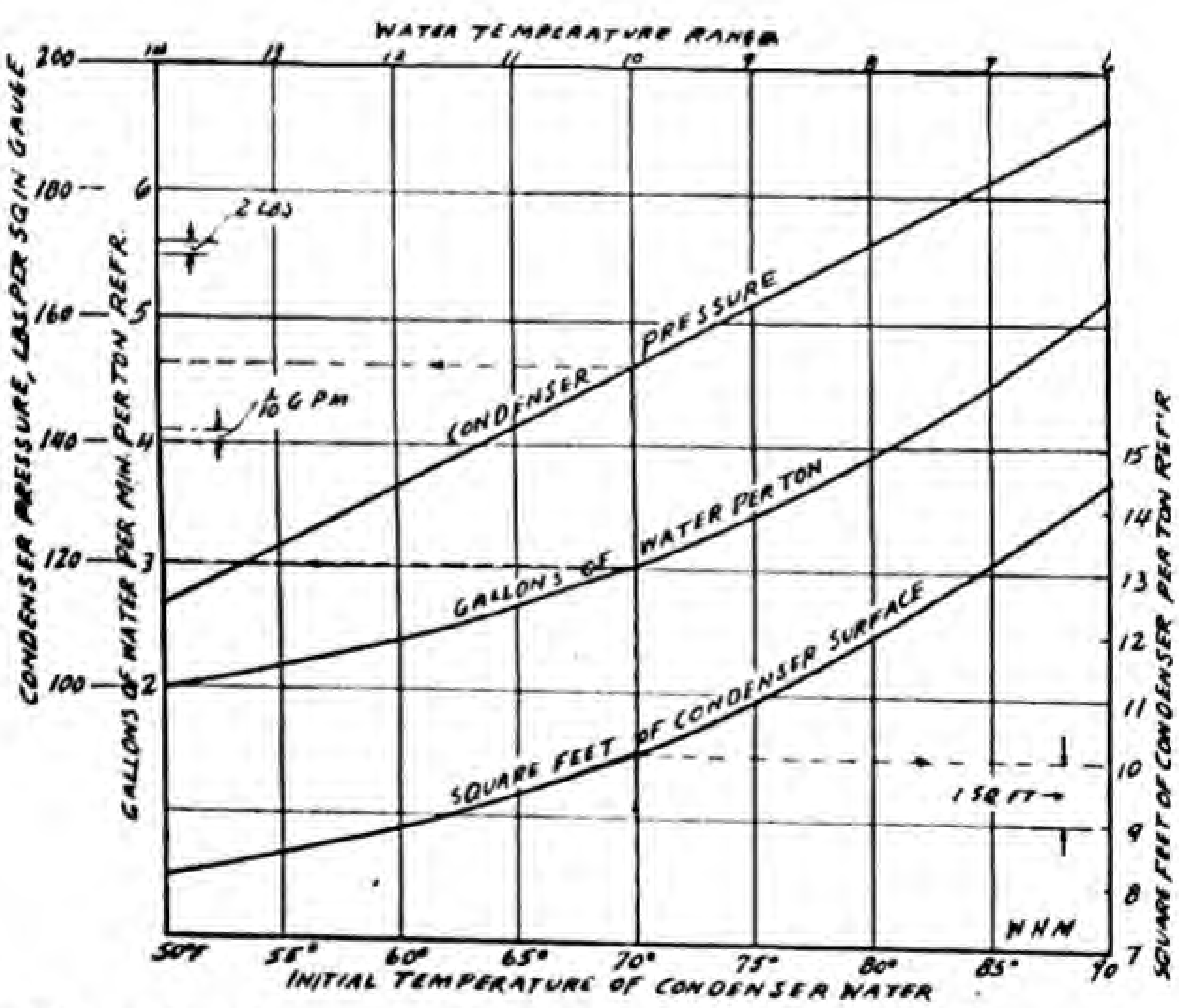


Fig. 33.—Operating Characteristics of Single-Pass Shell-and-Tube Condenser.

From the foregoing, it will be observed that the magnitude of the condenser pressure will depend upon three variables: The initial temperature of the condenser water, the quantity of the condenser water, and the magnitude of the condenser surface. It is evident that if the condenser surface and water are not supplied in sufficient quantities, the temperature difference must be correspondingly larger. This larger

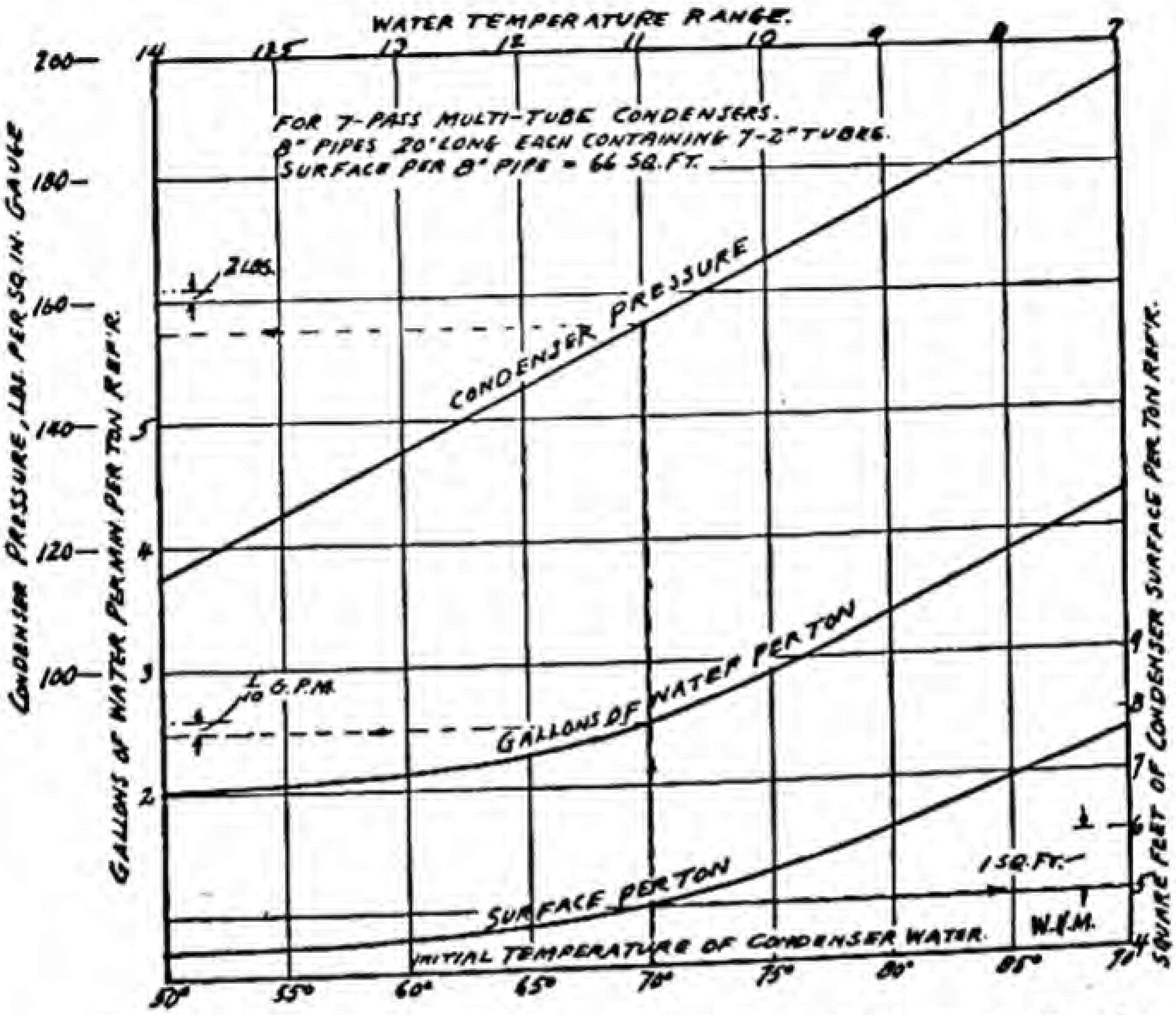


Fig. 34.—Operating Characteristics of 7-Pass Multitube Condensers.

temperature difference between the average temperature of the refrigerant and the average temperature of the water means that the condenser pressure must be correspondingly high. Due to the fact that the temperature of the condensing refrigerant must be always a few degrees above the temperature of the water, it is evident that the increase of the temperature of the water will, in a similar manner, increase the average temperature of the refrigerant,, which in turn means a higher condenser pressure.

Diagram of Cycle.—Due to the fact that the pressure heat-content diagram is destined to become a practical and useful diagram, Fig. 35 has been prepared to illustrate the use of this chart in a more graphical manner. Fig. 35 gives an outline of the original pressure heat-content diagram prepared by the Bureau of Standards, together with other lines which show the different cycles of operation.

For the purpose of illustrating and calculating the various quantities, it will be assumed that the saturated temperatures are 5° F. and 86° F. in the evaporator and condenser respectively. However, to illustrate the use of pressure heat-content diagram, the following thermodynamic laws must be held in mind: First, in a process which occurs at constant pressure, the increase in heat content is equal to the heat added; second, in adiabatic compression, the compression takes place at constant entropy, so that the theoretical equivalent of the work done by the compressor is equal to the change in the heat content of ammonia between the inlet and outlet to the compressor; third, in a throttling process, such as occurs at the expansion valve, the heat content of ammonia before and after passing through the expansion valve is the same (providing the velocity of ammonia is the same in both cases). In addition to the foregoing conditions, it will be further assumed that there is no transmission of heat between the ammonia and its surroundings, except in the condenser and evaporator unless, however, it is specifically mentioned that the liquid is aftercooled or that the saturated vapor is superheated. Also it is assumed that there is no friction in the system. With these limiting conditions, it is well to study the different phases of the different cycles of operation as indicated on Fig. 35.

Problems of Standard Conditions.—The pressures corresponding to the saturated temperatures of 5° F. and 86° F. in the evaporator and condenser may be graphically read from the saturated liquid line, which contain temperature at intervals of 2° F. Reading on this saturated liquid line at 5° F. and 86° F., the corresponding absolute pressure of the ammonia is seen to be 34.27 and 169.2 lbs. per sq. in. Starting with the saturated liquid ammonia coming from the condenser at 86° F., the state point representing this condition may be located on the diagram at the point where the 169.2 lbs. pressure line or 86° F. temperature line intersects the saturated liquid line. This is shown at the point A on Fig. 35.

In a similar manner, since the process in the expansion or throttling valve is one of constant heat content, the line representing this process will be vertical on this diagram, and is represented by the line AB, which line is determined by drawing a vertical line through the point A, until it intersects the 34.27-lb. pressure line or the 5° F. temperature line at right angles, at point B. The relation of point B in reference to the point K and C on the saturated liquid and saturated vapor lines determines the amount of liquid ammonia which has been evaporated to cool the ammonia from 86° to 5° F. This may be read from the diagram at point B, by reference to the constant quality curves on the

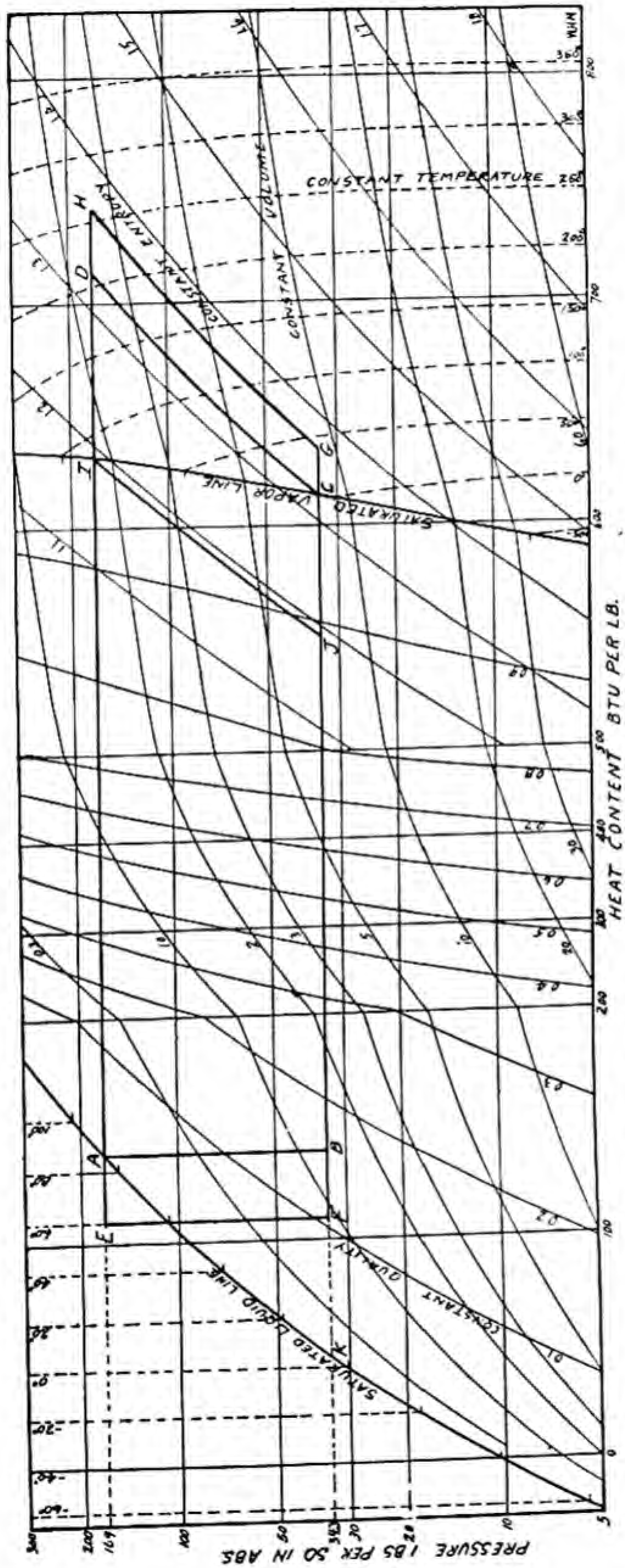


Fig. 35.—Heat Content Pressure Diagram for Ammonia.

Bureau of Standards pressure heat-content diagram. This is seen to be 0.160. This means that 16 per cent of the ammonia has been evaporated at the expansion valve. The quality of the mixture may be calculated by noting that the percentage of vapor in the mixture at the evaporator is proportionate to KB/KC. The heat contents at points K, B, and C, are 48.3, 138.9 and 613.3 Btu. respectively. The calculation of the amount evaporated is, therefore, as follows:

$$\frac{138.9-48.3}{613.3-48.3}=0.1603$$

When the heat is applied to the ammonia mixture in the evaporator, it evaporates at constant pressure until the mixture is changed to saturated vapor, or superheated vapor. If the ammonia is changed to saturated vapor, the process of evaporation in the evaporator will be represented by line BC in Fig. 35.

From the pressure heat-content diagram, the heat content of the ammonia vapor at 5° F. temperature or 34.3 lbs. as represented by point C on Fig. 35 is equal to 613.4 Btu. per lb. The amount of heat removed in the evaporator is, therefore, equal to the difference of heat contents at points B and C, which in this case, is equal to $613.3 - 138.9 = 474.4$ Btu.

The specific volume of the saturated vapor at 5° F. is seen to be near 8.2 cu. ft. per lb. from the chart. If the saturated vapor at 5° F. is compressed adiabatically, to a pressure of 169.2 lbs. corresponding to the saturated temperature of 86° F., the compression will be represented by line CD, which must coincide with a constant entropy line (which may be interpolated) through point C. The point D, lying on the 169.2 lbs. pressure line, represents the condition of the superheated vapor after adiabatic compression.

From the Bureau of Standards pressure heat-content diagram, the following conditions may be determined: specific volume, 2.36 cu. ft. per lb.; heat content, 712.9 Btu. per lb.; temperature, 210° F.

The heat equivalent of the adiabatic work of compression is represented by the difference of heat content at points C and D, which, in this case, is equal to $712.9 - 613.3 = 99.6$, Btu. per lb.

When the superheated ammonia vapor passes through the condenser it is cooled at constant pressure until the condensing (saturated) temperature is reached. This process of cooling is represented by lines DI in Fig. 35. The process of the condensing ammonia in the condenser is represented by line IA. Consequently, the heat removed in the condenser will be the difference in heat contents of ammonia at points D and A which, and in this case, is $712.9 - 138.9 = 574$ Btu. per lb.

It should be noted that the heat absorbed in the evaporator and heat added in the compressor will be equal to the heat rejected in the ammonia condenser; in this case, this is equal to $99.6 + 474.4 = 574$ Btu. per lb.

From the foregoing, it will be noted that the diagram *ABCDA* on Fig. 35 represents the dry compression refrigeration cycle under the assumed conditions and when there is no aftercooling of liquid ammonia, and when the suction vapor comes to the compressor in a saturated state.

Calculation of Additional Constants.—With the foregoing values of fundamental constants, additional and important factors may be calculated. Since a ton of refrigeration is the removal of heat at the rate of 200 Btu. per min., the amount of ammonia to be evaporated per min. to absorb heat at this rate may be calculated as follows:

$$\text{Lbs. per min. per ton} = \frac{200}{474.4} = 0.4216$$

When the compressor volumetric efficiency is 100 per cent, the theoretical amount of piston displacement required to remove the ammonia vapor at the rate of one ton of refrigeration per day is found as follows:

$$\text{Theoretical displacement} = 8.2 \times 0.4216 = 3.457 \text{ cu. ft. per min.}$$

The theoretical amount of power required per ton of refrigeration may be determined by ascertaining the heat equivalent of the work of compression. The heat equivalent of one hp. per min. is equal to 42.44 Btu. per min. The theoretical hp. per ton of refrigeration is therefore calculated as follows:

$$\text{Theoretical hp.} = \frac{99.6 \times 0.4216}{42.44} = 0.989$$

The theoretical coefficient of performance may be obtained by noting the ratio of heat absorbed in the evaporator, to the heat equivalent of the work of compression.

The theoretical amount of heat required per min. per ton of refrigeration for adiabatic compression is equal to 99.6×0.4216. The coefficient of performance would therefore be calculated as follows:

$$\text{Coefficient of performance} = \frac{200}{0.4216 \times 99.6} = 4.763$$

Other Cycles of Operation.—In addition to the foregoing cycles of dry compression with no superheating or aftercooling, the following additional cycles of operation may be noted:

1. Dry compression cycle with liquid ammonia aftercooled to 60° F. as it comes from the condenser.

2. Dry compression with the suction vapor superheated to 40° F.

3. Dry compression with both aftercooling and superheating effects.

4. Wet compression cycle with no aftercooling.

In the first case, the cooling of liquid ammonia from the saturated temperature of 86° F. to a temperature of 60° F. would be shown on the pressure heat-content diagram by a horizontal movement of the state point along the 169.2 lbs. pressure line. This cooling is denoted in Fig. 35 by the line *AE*. The process of expansion, through the expansion and throttling valve, being a process of constant heat content, is represented, therefore, by the vertical line drawn through *E* until it intersects 34.27-lb. pressure line at right angles at *F*. The cycle of operation with dry compression, and no superheating of the suction vapor but with aftercooling would, therefore, be represented by the diagram *EFCDE*.

In the event that the vapor coming from the evaporator is heated before it reachès the compressor to a temperature of 40° F., this process should be denoted by a horizontal movement of the state point along the 34.27 pressure line until the 40° F. temperature line is intersected, as at point *G* in Fig. 35. The cycle of operation with no liquid aftercooling, but with superheated suction vapor would be, therefore, shown by *ABGHA*. In the event that both liquid aftercooling to 60° F. and superheating to 40° F. occurs at the same time, the cycles of operation would be shown by the diagram *EFGHE* in Fig. 35.

In wet compression cycle, enough liquid ammonia is taken into the compressor to absorb the heat generated by the compression of the vapor. Theoretically, just enough liquid should be taken into the compressor so that the vapor should be in the saturated state at the end of the compression. However, under practical conditions a few degrees of superheating are required. Assuming that it is desired to maiñtain the compressed ammonia vapor in the saturated state at the end of compression, the quality of the mixture required for this condition should be obtained by following a constant entropy line through the point *I* until it intersects the 34.27-lb. pressure line at point *J* on Fig. 35. The quality of the mixture is represented by point *J* and may be obtained directly from the Bureau of Standards' pressure heat-content diagram, which, in this case, is 0.889. The cycle of operation for wet compression with no liquid aftercooling under these conditions should be represented by diagram *ABJIA*, on Fig. 35.

All of the fundamental constants for all of the foregoing cycles of operation have been taken from the Bureau of Standards' pressure

TABLE 31.—FUNDAMENTAL CONSTANTS.

	Dry compression with no aftercooling and no superheating	Dry compression with liquid cooled to 60°	Dry compression with vapor superheated to 40°	Dry compression with liquid aftercooled to 60° and vapor heated to 40°	Wet compression with no aftercooling or superheating
1. Temperature in evaporator	5°	5°	5°	5°	5°
2. Pressure in evaporator	34.3	34.3	34.3	34.3	34.3
3. Specific volume of vapor, to comp.	8.2	8.2	8.9	8.9	7.24
4. Temperature in condenser	86°	86°	86°	86°	86°
5. Pressure in condenser	169	169	169	169	169
6. Temperature after compression	210°	210°	259°	259°	86°
7. Specific volume of gas after compression, cu. ft.	2.36	2.36	2.58	2.58	1.77
8. Heat content of superheated vapor, B.t.u. per lb.	712.9	712.9	742.5	742.5	631.5
9. Heat content of vapor from evaporator, B.t.u.	613.4	613.4	634.0	634.0	550.5
10. Heat equivalent of work of compression, B.t.u.	99.5	99.5	108.5	108.5	81.0
11. Heat content of liquid in condenser, B.t.u.	138.9	109.3	138.9	109.3	138.9
12. Heat rejected in condenser (item 8—item 11), B.t.u.	574	603.6	603.6	633.2	492.6
13. Refrigerating effect (item 9—item 11), B.t.u.	474.5	504.1	495.1	524.7	411.6
14. Quality of mixture after expansion	0.160	0.108	0.160	0.108	0.160
15. Lbs. of ammonia per minute per ton of refrigeration	0.4216	0.3970	0.4040	0.3811	0.4862
16. Theoretical volume of ammonia per ton per minute, cu. ft.	3.457	3.257	3.596	3.392	3.52
17. Theoretical horsepower per ton	0.988	0.931	1.033	0.974	0.928
18. Coefficients of performance	4.768	5.070	4.563	4.841	5.080
19. Quality of mixture before compression					0.890

heat-content diagram, as shown in Fig. 35. The calculated results of some of the other fundamental constants are shown also in Table 31.

All of the foregoing cycles of operation have been shown in Fig. 35 for the operating temperature of 5° F. and 86° F. in the evaporator and condenser respectively. Other cycles of operation for any operating pressures between 5 and 300 lbs. per sq. in. abs. pressure may be shown in a manner similar to Fig. 35.

Diagram of System.—Fig. 36, is a diagram of the ammonia compression refrigerating system suggested by the author to his brother, O. W. Motz, who incorporated it into his thesis for B. S. degree in mechanical engineering, at the Rose Polytechnic Institute, Terre Haute, Ind., in June, 1925. This diagram gives a good picture of the changes of temperature, pressure, volume, heat content and entropy in a complete compression system, during a complete cycle of operations. It is based on the following propositions:

1. Saturated condenser temperature = 86° F.
2. Saturated evaporator temperature = 5° F.
3. Refrigerator temperature = 25° F.
4. Water temperature rise in condenser 80 − 70 = 10° F.
5. In the constant pressure processes in the condenser and evaporator, the heat change is equal to the change in heat content.
6. In an adiabatic compression, the heat equivalent of the work compression is equal to the change of heat content of refrigerant between inlet and outlet.
7. In a throttling process, such as occurs at the expansion valve, the heat content is constant.
8. There is no loss by friction or radiation.

In Fig. 36, the parts of the compression system are shown in diagrammatic form. The diagrams for the evaporator and condenser show how the volumes of the liquid and vapor vary.

In Fig. 36, the lines and curves have the following meanings:

Line A, B, C, D, E, F, G represents change of temperature.
Line H, I, J, K, L represents change of absolute pressure.
Line M, N, O, P, Q, R, S, T represents change of specific volume.
Line U, V, W, X, Y, Z represents change of heat content.
Line abcde represents change of entropy.

Throttling of Liquid.—Due to the fact that the pressure in the evaporator is always below that in the condenser, the liquid refrigerant coming from the liquid storage tank or receiver must be throttled. The liquid may be at the saturation temperature corresponding to the condenser pressure, but its pressure drop or amount of throttling is the same in either case. The temperature of the liquid refrigerant is reduced from that existing before the valve or throttle device to the saturated temperature corresponding to the pressure in the evaporator.

Example.—Saturated liquid ammonia is throttled from absolute pressure of 169.2 lbs. to 34.27 lbs. What is the temperature drop?

SOLUTION—Saturated temperature at 169.2 lbs. = 86° F.
Saturated temperature at 34.27 lbs. = 5° F.

Temperature drop........... = 81° F.

Example.—Ammonia condenses at a saturated temperature of 86° F.; evaporating temperature is 5° F., what is pressure drop in expansion valve?

SOLUTION—Absolute pressure at 86° sat. temp. = 169.20 lbs.
Absolute pressure at 5° sat. temp. = 34.27 lbs.

Total drop of pressure....... = 134.93 lbs.

The above temperature drop of 81° F. and pressure drop of 134.93 lbs. are represented by lines AB and HI of Fig. 36.

If the liquid ammonia comes to the expansion valve at 86° F. its heat content is 138.9 Btu. per lb., but the liquid must be cooled to 5° F. To do this some of liquid is evaporated. The heat content of liquid ammonia at 5° F. is 48.3 Btu. per lb. so that the difference, 138.9 − 48.3 = 90.6 Btu. must be supplied to cool the liquid. This heat, 90.6 Btu., is obtained by evaporating some of the liquid. At 5° F. evaporating temperature the latent heat of ammonia is 565 Btu. per lb. Then it is evident that the ratio of 90.6 ÷ 565 = 0.1603 = 16.03 per cent represents the amount of the pound that must be evaporated to cool the liquid. It should be noted that the total heat in a pound is not changed in this process, and that only some of heat of liquid is interchanged with equivalent latent heat of evaporation. Since there is no change of heat content, point U in Fig. 36, represents the heat content of the ammonia in the expansion valve.

The volume of 1 lb. of liquid at 86° F. is 0.0269 cu. ft. and is represented by point M in Fig. 36, after passing through the valve or throttle device, the mixture is 16.03 per cent gas and 100.00 − 16.03 = 83.97 per cent liquid. The total volume of the pound of mixture is the sum of the above portions of liquid and vapor. The specific volume of 1 lb. of vapor at 5° F. is 8.15 cu. ft. The volume of 1 lb. of mixture is found as follows, remembering that the volume of 1 lb. of liquid at 5° F. is 0.0243 cu. ft.:

Volume of liquid = 0.0234 × 0.8397 = 0.0204
Volume of vapor = 8.15 × 0.1603 = 1.3064

Total volume, cu. ft. per lb. ...... = 1.3268

The total volume per lb. of mixture is 1.3268 cu. ft. is represented by point N of Fig. 36.

The entropy of liquid ammonia at 86° F., from tables is 0.2875, and is represented by point *a* of Fig. 36. The entropy of liquid ammonia at 5° F. is 0.1092 and the entropy of the vapor is 1.3253. The entropy of the mixture after the expansion valve is found as follows:

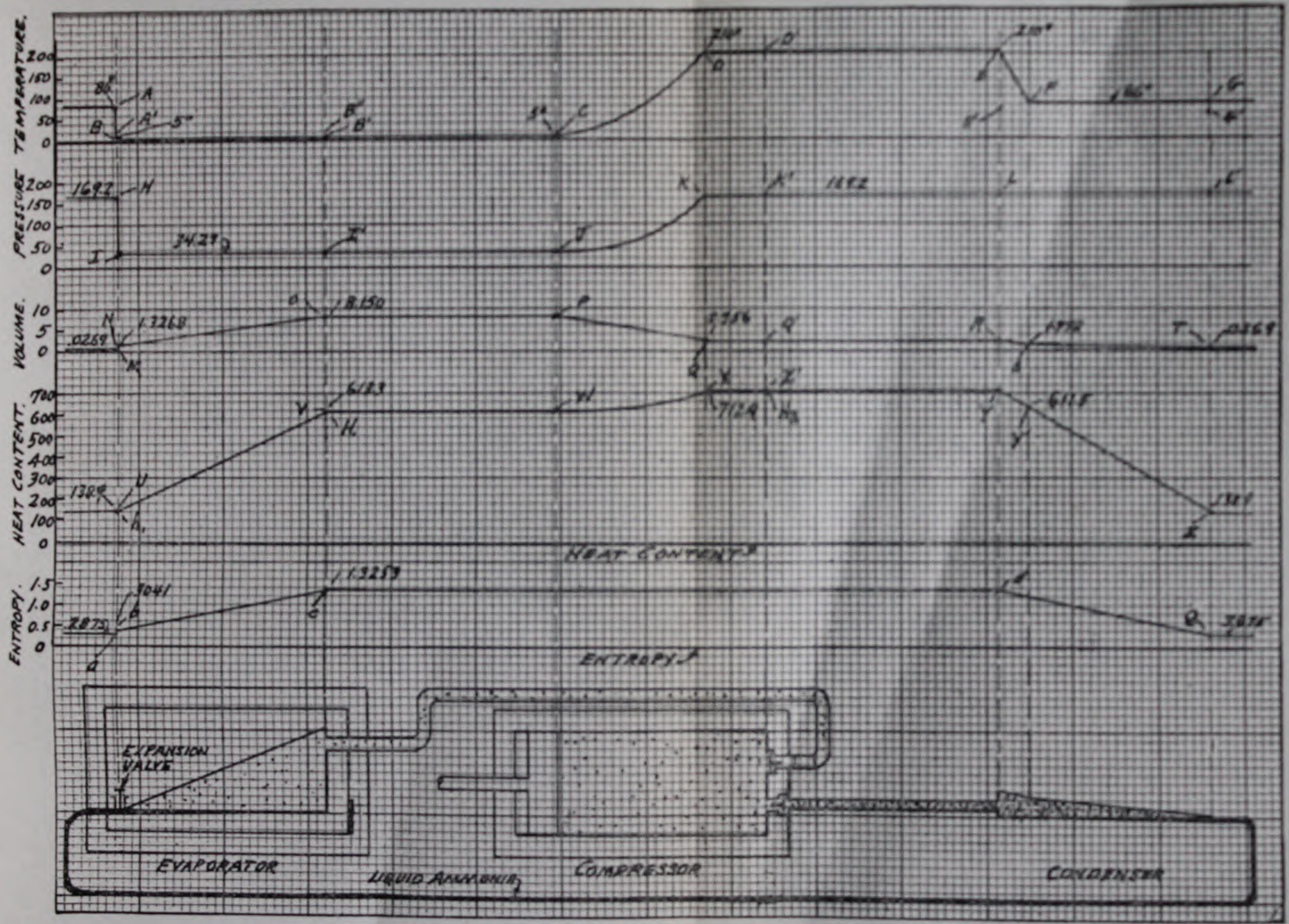


Fig. 36.—Diagram of the Ammonia Compression Refrigerating System.

$$\begin{aligned}
\text{Entropy of liquid} &= 0.8397 \times 0.1092 = 0.0917 \\
\text{Entropy of vapor} &= 0.1603 \times 1.3253 = 0.2124 \\
\text{Total entropy of mixture} \ldots\ldots\ldots &= 0.3041
\end{aligned}$$

The entropy of 0.3041 of the mixture after the expansion is represented by point *b* of Fig. 36.

Evaporation of Liquid Refrigerant.—Neglecting friction and other losses, the evaporation of the refrigerant takes place at constant pressure and temperature. The relative temperature of evaporation will depend upon the pressure maintained in the evaporator. Thus for ammonia evaporating at 5° F. the absolute pressure would be 34.27 lbs. per sq. in.

The line BB′ of Fig. 36, represents the temperature in the evaporator for ammonia at 5° F. The line II′ represents the corresponding pressure. The line A′B″ represents a constant refrigerator temperature of 25° F.

If the ammonia vapor leaves the evaporator in a dry and saturated state its heat content would be 613.3 Btu. per lb. or represented by point V of Fig. 36. If the liquid contains no vapor and is at a temperature of 86° F. at the expansion valve, its heat content is 138.9 Btu. per lb. Evidently the heat absorbed in the evaporator is 613.3 − 138.9 = 474.4 Btu. per lb. This process is represented by line UV of Fig. 36.

If the ammonia vapor leaves the evaporator in the dry and saturated state, its volume would be 8.15 cu. ft. per lb. under the volume increases in the evaporator from 1.3268 to 8.15, which process is presented by line NO of Fig. 36.

Compression of Vapor.—Since the compression is adiabatic and since entropy may be defined as ratio of the actual heat added to the absolute temperature, it is evident that the entropy of the gas will remain constant during the compression of the gas. This fact, together with the data in the U. S. Bureau of Standards Ammonia Tables, makes it possible to determine the condition of the gas at any point in the compression stroke of the compressor. From the tables it will be noted that the entropy of vapor at 5° F. and 34.27 lbs. is 1.3253. Thus by referring to the superheat ammonia tables it is possible to obtain from these the temperature, heat content and specific volume at any point of compression stroke. Thus, by using adiabatic compression with a constant entropy of 1.3253, it is possible to obtain the following values of temperature, heat content, specific volume, and stroke volume for the various compression pressures as shown in Table 32.

Columns 1, 2, 3 and 4 may be obtained from the superheat ammonia table at a constant entropy of 1.3253. Column 5, giving the percentage of stroke completed for the pressures shown in column 1, may be calcu-

TABLE 32.—CONDITION OF AMMONIA DURING COMPRESSION.
By O. W. MOTZ

Absolute pressure lbs. per sq. in. 1	Temperature of gas deg. F. 2	Heat content Btu. per lb. 3	Specific volume cu. ft. per lb. 4	Percentage of stroke completed 5
34.27	5.0	613.3	8.150	0.00
50	47.2	633.7	6.094	25.24
75	96.9	657.8	4.455	45.34
100	134.7	676.1	3.562	56.32
125	165.7	691.2	2.991	63.30
150	191.9	704.0	2.592	68.20
169.2	209.7	712.9	2.358	71.09

lated as follows: The compressor cylinder may be considered to have a volume of 8.15 cu. ft. which corresponds to the specific volume of the intake ammonia. From the tables it is found that specific volume when the pressure reaches 169.2 lbs. is 2.358, consequently the percentage of stroke completed at this point is calculated as follows:

$$(8.150 - 2.358) \div 8.150 = 0.7109 = 71.09\%.$$

The foregoing data are shown graphically by Fig. 36:

Line CDD′ shows temperatures during compression.
Line JKK′ shows pressures during compression.
Line PQQ′ shows specific volumes during compression.
Line WXX′ shows heat contents during compression.
Line cd shows entropy during compression.

Condensation of Vapor.—For the 5° F. and 86° F. standard conditions it was previously shown that the vapor leaves the compressor at a temperature of 209.7° F., due to theoretical adiabatic compression. This is the temperature of the vapor at entrance to the condenser. If the condenser has sufficient surface and water which rises from 70° to 80° F. in passing through it, the condenser temperature may be 86° F., with a corresponding condensing pressure of 169.2 lbs. as for ammonia. Neglecting the friction of the condenser, the pressure will be constant in all parts of it. This is shown by line LL′ of Fig. 36.

In the first part of the condenser the gas is cooled from 209.7° to 86° F., which is the removal of the superheat of the ammonia. This process is represented by line EF of Fig. 36. When the ammonia reaches the temperature of 86° F. it will begin to condense, and will remain at a constant temperature until the vapor is entirely condensed. This action is represented by line FG. In passing through the condenser, the water is heated from 70° to 80° F. This is shown by line E′G′.

In passing through the condenser the volume of the refrigerant decreases rapidly, since it comes in as a superheated vapor and leaves as a liquid. For ammonia under the standard conditions it has been pre-

viously shown that the specific volume of the gas as discharged by the compressor is 2.356 cu. ft. per lb. At a condensing temperature of 86° F., the specific volume is found to be 1.772 from the ammonia tables also, if the liquid leaves the condenser at 86° F., its specific volume is found to be 0.0269 from the tables. This change of volume of the ammonia during condensation is shown by line RST of Fig. 36.

Since the gas enters the condenser at a temperature of 209.7° F. and at a pressure of 169.2 lbs., the heat content is 712.9 Btu. per lb. If the gas is cooled at a constant pressure of 169.2 lbs. to a temperature of 86° F. its heat content will be 631.5 Btu. The difference, 712.9 − 631.5 = 81.4 Btu., is the superheat in the gas for the conditions stated. Now, if the vapor is entirely condensed and is at a pressure of 169.2 lbs. and temperature of 86° F., the heat content of the liquid will be 138.9 Btu. per lb. The difference, 631.5 − 138.9 = 492.6 Btu., represents the latent heat of condensation.

The line YY'Z represents the foregoing change of heat content of the ammonia as it passes through the condenser under the stated conditions.

It has been previously shown that the entropy of the gas leaving the compressor is 1.3253. This is the entropy of the gas as it enters the condenser, and is represented by point d of Fig. 36. From the ammonia tables it may be found that the entropy of the liquid leaving the condenser at 86° F. is 0.2875. This is represented by point e. The line de therefore represents the change of entropy of the ammonia as it passes through the condenser under the assumed conditions.

Compound Ammonia Compression.—In previous years, it has been considered economical to install the ammonia absorption refrigerating machine when the temperatures in the cold storage plant were such that the suction pressure must be carried below 10 lbs. per sq. in. gauge. It is obvious that many single-stage ammonia compression plants are installed which operate at a lower suction pressure, but for economical operation, as previously indicated, the ammonia absorption or compound compression system should be used. The two-stage ammonia compression system is somewhat more complicated than the simple compression system, and consists of the following major parts of apparatus: Low-pressure compressor cylinder, high-pressure compressor cylinder, low-pressure discharge vapor cooler, and an intermediate receiver. The low-pressure compressor cylinder draws in the ammonia vapor from the evaporator. It is compressed to a medium pressure in the low-pressure cylinder and is discharged into a gas cooler. This is generally cooled by means of water and is used for the purpose of removing the superheat of the ammonia gas. In low temperature work the desuperheating and cooling of the liquid refrigerant may be done by direct expansion since both are done at low temperature levels.

TABLE 33.—TEST OF COMPOUND AMMONIA COMPRESSION PLANT.

Pressures and Temperature °F — Ammonia

Time	Pressures: Disch. at Machine H. P.	Pressures: Disch. at Machine Int. P.	Pressures: Suct., Mercury Column, Inches	Pressures: Barometer	Ammonia Liquid: From Receiver	Ammonia Liquid: From Liquid Cooler	Ammonia Liquid: To Brine Cooler	Ammonia Gas: At Cond's Inlet	Gas at Machine: L. P. Suct.	Gas at Machine: L. P. Disch.	Gas at Machine: H. P. Suct.	Gas at Machine: H. P. Disch.	Gas: From Int. Gas Cooler
9:30	170	70	16½		88.1	75.4	48.1	147.0	−8.0	182	51	172	100.0
9:45	*	*	*		80.0	75.9	*	148.0	*	*	*	174	*
10:00	171	*	16⅝	30.10	87.2	76.0	*	149.0	*	*	*	173	*
10:15	*	*	*		87.7	76.3	48.0	150.0	*	*	*	174	*
10:30	172	*	*		90.1	76.5	48.1	150.0	−8.0	*	*	175	*
10:45	*	*	16¾		90.3	76.9	48.3	152.0	*	*	*	177	100.5
11:00	173	*	16⅝		91.1	77.1	*	*	−7.0	183	*	*	*
11:15	*	*	16⅝		91.1	77.3	*	153.0	*	182	*	*	*
11:30	*	*	16⅝		91.4	74.4	48.1	154.0	−8.0	*	*	*	100.0
11:45	*	*	*		91.2	*	48.0	*	*	*	*	178	*
12:00	*	71	*	30.08	*	77.5	*		−9.0	*	*	*	*
12:15	*	*	16		*	*	*		−9.0	181	*	*	*
12:30	*	*	*		91.8	77.8	*	156.0	−10.0	180	*	*	*
12:45	174	72	15⅞		91.5	77.5	*	155.5	*	179	50	*	99.5
1:00	176	73	*	30.06	92.2	77.9	*	156.0	*	*	51	180	*
1:15	180	*	15 11/16		92.7	78.2	48.3	156.0	−11.0	180	*	181	*
1:30	*	*	15⅞		93.1	78.6	*	157.5	*	*	*	183	*
1:45	*	*	15¾		*	*	48.0	158.0	*	179	*	181	*
2:00	*	*	15⅝		93.5	78.8	48.2	158.5	*	180	*	184	99.0
2:15	183	*	*	30.05	94.0	79.0	48.3	160.0	*	*	*	185	*
2:30	*	*	*		94.8	79.6	48.4	*	*	181	*	187	*
2:45	*	74	*		94.8	80.6	48.5	161.0	*	182	*	189	100.0
3:00	*	73	15½	30.05	93.4	79.4	48.0	*	*	180	*	184	99.5
3:15	*	*	*		*	78.9	*	*	*	181	50	*	99.0
3:30	180	72	*		92.9	79.4	47.8	*	−10.0	180	*	*	*
3:45	*	*	15½		92.2	78.8	47.5	160.0	*	*	*	181	*
4:00	176	*	15⅝	30.06	*	78.2	*	*	*	*	*		*
4:15	178	*	*		92.9	78.6	47.8	*	−10.0	181	*	184	*
4:30	179	*	15⅝		93.2	78.8	48.0	*	*	182	*	186	*
4:45	183	*	15½		94.1	79.1	48.4	*		183	51	*	100.0
5:00	*	73	*		94.2	79.2	48.5	161.0		*	*	*	*
5:15	*	*	*		94.3	79.3	48.4	162.0	−9.0	*	*	*	*
Avg.	177	71.7	15.8	30.08	92.18	78.24	48.1	156½	−9.3	181.2	50.8	181	99.7

Temperature °F — Brine and Water

Time	Brine: In	Brine: Out	Brine Range: Mercury Therm.	Brine Range: Elect. Resistance Therm.	Water through Condensers: On	Water through Condensers: Off	Water through Condensers: Range	Through Liquid Cooler: On	Through Liquid Cooler: Off	Through Liquid Cooler: Range	Through Interm. Gas Cooler: On	Through Interm. Gas Cooler: Off	Through Interm. Gas Cooler: Range
9:30	1.63	−3.55	5.16	5.20	60.4	84.1		58.9	67.0	8.1	67.3	88.7	21.4
9:45	1.63	*	5.18	*	60.7	83.8		59.0	67.2	8.2	67.4	88.9	21.5
10:00	1.63	*	5.18	*	61.1	84.2		58.9	67.2	8.3	67.7	89.0	21.3
10:15	1.63	−3.57	5.20	5.23	62.5	84.6		59.1	67.4	*	67.8	*	21.2
10:30	1.50	−3.60	5.10	5.15	63.4	85.6		58.9	67.6	8.7	67.9	*	21.1
10:45	1.42	−3.70	5.12	*	62.7	85.6		59.0	67.7	*	68.0	*	21.0
11:00	1.40	−3.72	*	*	63.2	85.8		*	*	*	*	*	*
11:15	1.37	−3.77	5.14	*	63.1	86.0		*	67.4	8.4	*	*	*
11:30	1.30	−3.90	5.20	5.18	62.7	*		*	67.5	8.5	*	88.8	20.8
11:45	1.20	−3.93	5.13	5.17	63.1	85.9		*	67.4	8.4	*	*	*
12:00	1.05	−4.02	5.07	5.10	63.5 63.1	86.0 85.9		59.1	67.5	*		88.7	20.7
12:15	1.00	−4.03	5.06	*	63.3 63.6	85.8 86.2		59.0	67.4	*	*	88.6	20.6
12:30	1.00	*	*	5.03	62.3 64.0	85.9 85.8		*	67.2	8.2	*	88.4	20.4
12:45	0.90	−4.13	5.03	5.05	64.5 64.7	86.6 86.7			67.1	8.1	67.6	88.0	*
1:00	0.80	−4.22	5.02	*	65.7 65.2	87.1 87.5		58.9	67.3	8.4	67.8	*	20.2
1:15	0.80	*	*	5.07	65.8 65.6	87.5 88.0		59.0	67.5	8.5	68.0	88	20.0
1:30	0.78	−4.25	5.03	*	65.4 64.8	* 87.6		*	67.3	8.3	*	*	*
1:45	0.72	−4.33	5.06	5.10	66.2 67.2	* 88.1		*	67.1	8.1	67.8	87.8	*
2:00	0.70	−4.38	5.08	*	67.2 66.5	88.5 87.7		59.1	67.3	8.2	67.9	*	19.9
2:15	0.70	−4.40	5.10	5.11	68.8 67.5	89.1 89.8		59.0	67.4	8.4	68.0	88.0	20.0
2:30	0.85	*	5.06	5.08	68.5 68.8	93.0 94.0		*	67.8	8.8	*	*	*
2:45	0.83	*	5.03	5.06	69.0 68.3	88.0 87.3		*	67.4	8.4	68.5	88.3	19.8
3:00	0.83	−4.45	5.08	5.10	69.8 70.4	87.0 87.2		*	66.9	7.9	67.5	88.0	20.5
3:15	0.60	*	5.05	5.08	70.0 69.8	87.8 87.1		*	67.3	8.3	67.7	87.7	20.0
3:30	0.80	−4.40	5.00	5.04	68.5 67.9	86.4 86.0		*	66.7	7.7	*	87.8	20.1
3:45	0.60	−4.45	5.05	5.09	68.0 70.1	85.8 85.9		*	66.5	7.5	67.0	87.5	20.5
4:00	0.70	−4.40	5.10	5.14	69.7 71.4	86.5 87.0		*	66.7	7.7	*	87.6	20.0
4:15	0.70	*	5.10	5.08	70.6 71.7	87.8 87.8		*	67.1	8.1	67.6	87.5	19.9
4:30	0.50	−4.50	5.00	5.07	71.1 *	88.4 88.3		*	67.5	8.5	67.9	87.9	20.0
4:45	0.55		5.03	5.02	71.4 70.8	88.5 *		59.1	68.1	9.0	68.7	88.6	19.9
5:00	0.55	*	*	*	71.6 71.3	* 88.3		*	68.2	9.1	68.5	88.9	20.1
5:15	0.50	*	5.00	*	70.9 70.5	88.3 87.8		*	68.2	*	68.8	89.0	20.2
Avg.	0.94	−4.13	5.07	5.09			20.72	59.01	67.35	8.34	67.88	88.34	20.44

Temperature °F — Air; Elect. Measurements; Quantities per Minute

Time	Air: Near Brine Cooler	Air: Near Inlet Gas Cooler	Air: Near Interm. Receiv.	Air: Between Compressors	Air: In Condenser Room	Indicated Kilowatts	Power Factor	Exciter A	Exciter V	Lbs. Liquid NH_3	Gals. Water Through: Condensers	Gals. Water Through: Inter Gas Cooler	Gals. Water Through: Liquid Cooler	Gal. Brine Through Brine Cooler
9:30	78.0	86	85.0	86.5	66.5	364	100	77	149	128	350	32.7 277#	32.7 277#	1605
9:45	79.0	*	84.0	*	66	*	*	*	*	127½	*	*	*	*
10:00	*	87	83.5	86.0	66.5	*	*	*	*	*	*	*	*	*
10:15	*	*	84.5	*	68.5	*	*	*	*	*	*	*	*	*
10:30	78.0	*	83.0	85.5	63	368	*	*	*	129	*	*	*	*
10:45	79.5	*	87.0	86.0	66.5	*	*	*	*	*	*	*	*	*
11:00	78.5	*	83.5	85.5	65.5	*	*	*	*	*	*	*	*	*
11:15	79.0	*	83.0	86.0	65.0	*	*	*	*	125	*	*	*	*
11:30	80.0	88	86.5	*	66.0	*	*	*	*	*	*	*	*	*
11:45	79.0	87	*	85.5	64.5	368	*	*	*	*	*	*	*	*
12:00	*	88	*	*	60.5	364	*	*	*	124½	*	*	*	*
12:15	*	*	83.0	*	68.0	*	*	*	*	*	*	*	*	*
12:30	79.5	*	82.5	*		*	*	*	*	120	*	*	*	*
12:45	78.0	87	83.0	86.5	63.5	*	*	*	*	119	*	*	*	*
1:00	79.0	88	*	86.0	66.0	366	*	*	*	121	*	*	*	*
1:15	*	*	*	*	67.0	*	*	*	*	122½	*	*		*
1:30	80.0	*	*	*	67.0	365	*	*	*	117	*	*	*	*
1:45	78.5	*	84.0	*	65.5	371	*	*		117½	*	*		*
2:00	80.0	*	83.0	86.5	67.0	*	*	*		*	*	*	*	*
2:15	79.0	*	*	85.5		376	*	*	*	121	*	*	*	1600
2:30	*	*	*	87.0	63.5	380	*	*	*	*	*	*	*	*
2:45	80.0	*	*	*	67.5	376	*	*	*	118	450	*	*	*
3:00	*	89	84.0	*	66.0	372	*	*	*	114	*	*	*	*
3:15	79.0	*	83.5	88.0	65.5	*	*	*	*	116	*	*	*	*
3:30	79.0	*	83.0	87.0	67.5	*	*	*	*	*	*	*	*	*
3:45	80.0	88	83.5	*	69.0	366	*	*	*	114	*		*	*
4:00	79.5	*	83.0	*	67.0	368	*	*	*	114	*	*	*	*
4:15	80.0	89	84.0	86.5	69.5	*	*	*	*	117	*	*	*	1605
4:30	81.0	*	83.0	87.0	*	372	*	*	*	125	*	*	*	*
4:45	81.0	*	84.0	*	*	368	*	*	*	*	*	*	*	*
5:00	80.0	*	*	*		*	*	*	*	*	*	*	*	*
5:15	80.0	*	84½	*	*	*	*	*	*	*	*	*	*	*
Avg.	79	88	83.4	86.3	67	368.3	100	77	149	122*	384.4	32.7	32.7	1604

No. 2, Machine in operation
No. Brine Coolers in operation, 2
Specific Gravity of Brine at 68° F = 1.2165, compared with water at 39.1° F
Specific Heat of Brine = 0.681

* Corrected for temperature, this figure becomes 123½

The ammonia vapor from the gas cooler is next led to an intermediate receiver. The liquid coming from the ammonia condenser, or the ammonia liquid cooler, is expanded into this intermediate receiver or trap. The vapor which is produced due to cooling the liquid from the temperature at the inlet to the temperature corresponding to the pressure in the gas cooler, unites with the vapor coming from the gas cooler and is drawn into the high pressure cylinder. The high pressure cylinder takes this vapor at the medium pressure and discharges it into the condenser against the condenser pressure.

The liquid from the bottom of the intermediate receiver or trap is led to the evaporator, and the pressure is reduced from the pressure in the intermediate trap and vapor cooler to that in the evaporator by means of an expansion or regulating valve. The principal advantages secured by compound ammonia compression are the reduction of horsepower required for compression, the reduction of the discharge temperature, etc.

Mr. George A. Horne of New York read a paper at the annual meeting of the American Society of Refrigerating Engineers in 1921, in which he described an elaborate test which was performed on a compound ammonia compression system in New York. The compressors were direct-connected synchronous motor-driven, having horizontal double-acting compressor cylinders. The test was performed upon a machine having a 22-in. diameter low-pressure cylinder and 13-in. diameter high-pressure cylinder, both cylinders having a stroke of 24 in. These compressors operated at a speed of 150 r.p.m. Fig. 37 is a diagrammatic reproduction of the connections in this plant. In Fig. 37 is also shown the temperatures at the various parts of the cycle during the test performed on October 17, 1921. The log of this test is shown by Table 33.

Another arrangement of two-stage compression is illustrated in Fig. 38. It will be seen that the two compressors are on the same shaft, like any cross-compound steam-engine, and as a rule the receiver (intermediate cooler) pressure is designed so as to provide equal work in each cylinder. The only time this is deviated from is when it is desired to operate with two suction pressures, in which case a receiver pressure of 20 to 25 lbs. is carried.

The gas discharge from the low pressure cylinder has a temperature of 120-180° F., depending on the pressure in the intercooler. The temperature to which this may be cooled by the available cooling water may vary from 55 to 90 or more degrees, which is not enough for economical operation. In consequence some other means of cooling must be obtained. Referring to the diagrammatical arrangement, it will be seen that a certain amount of the liquid ammonia from the liquid receiver passes first into a special trap accumulator, and any surplus

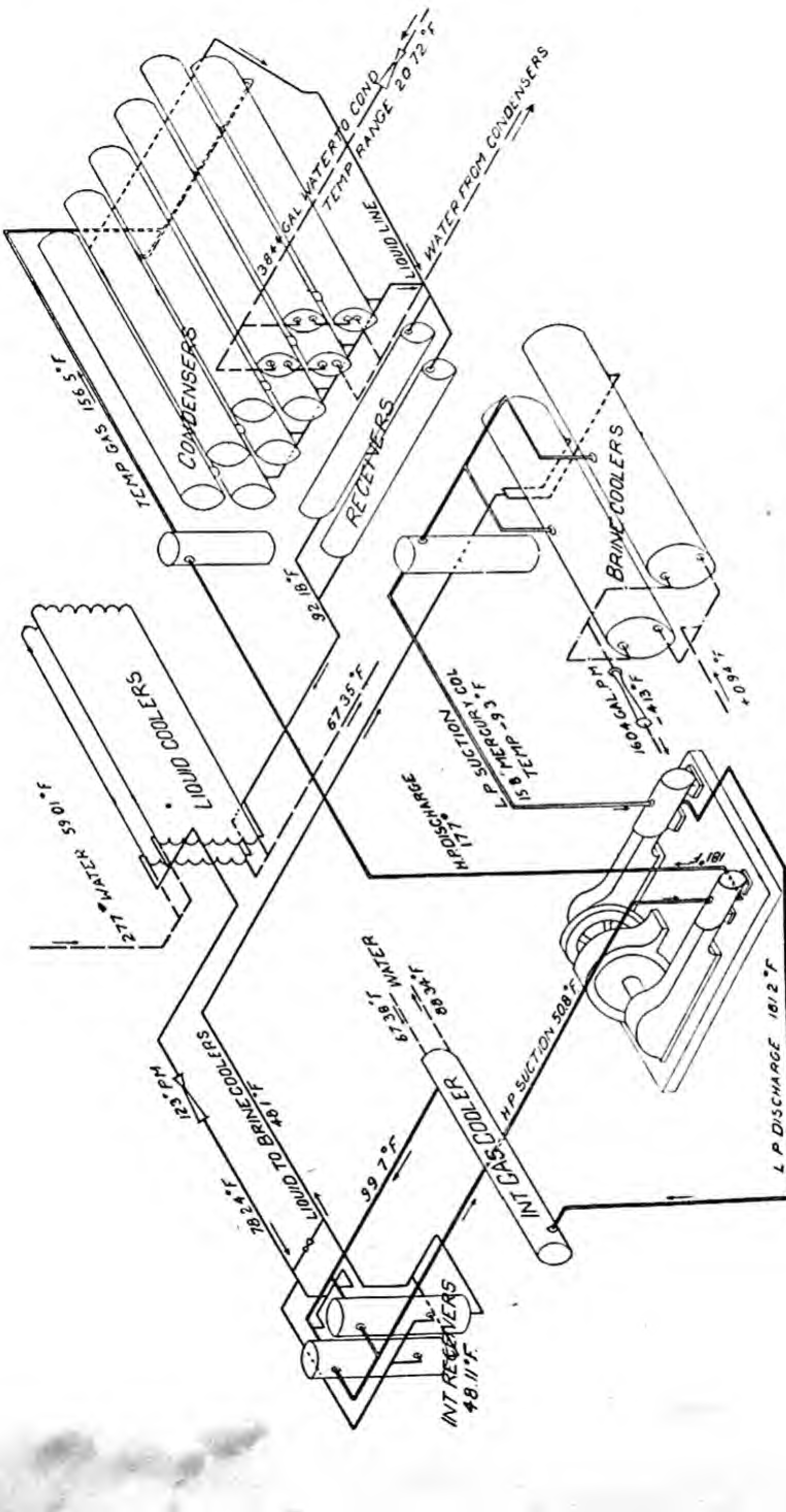


Fig. 37.—Compound Ammonia Compression Plant.

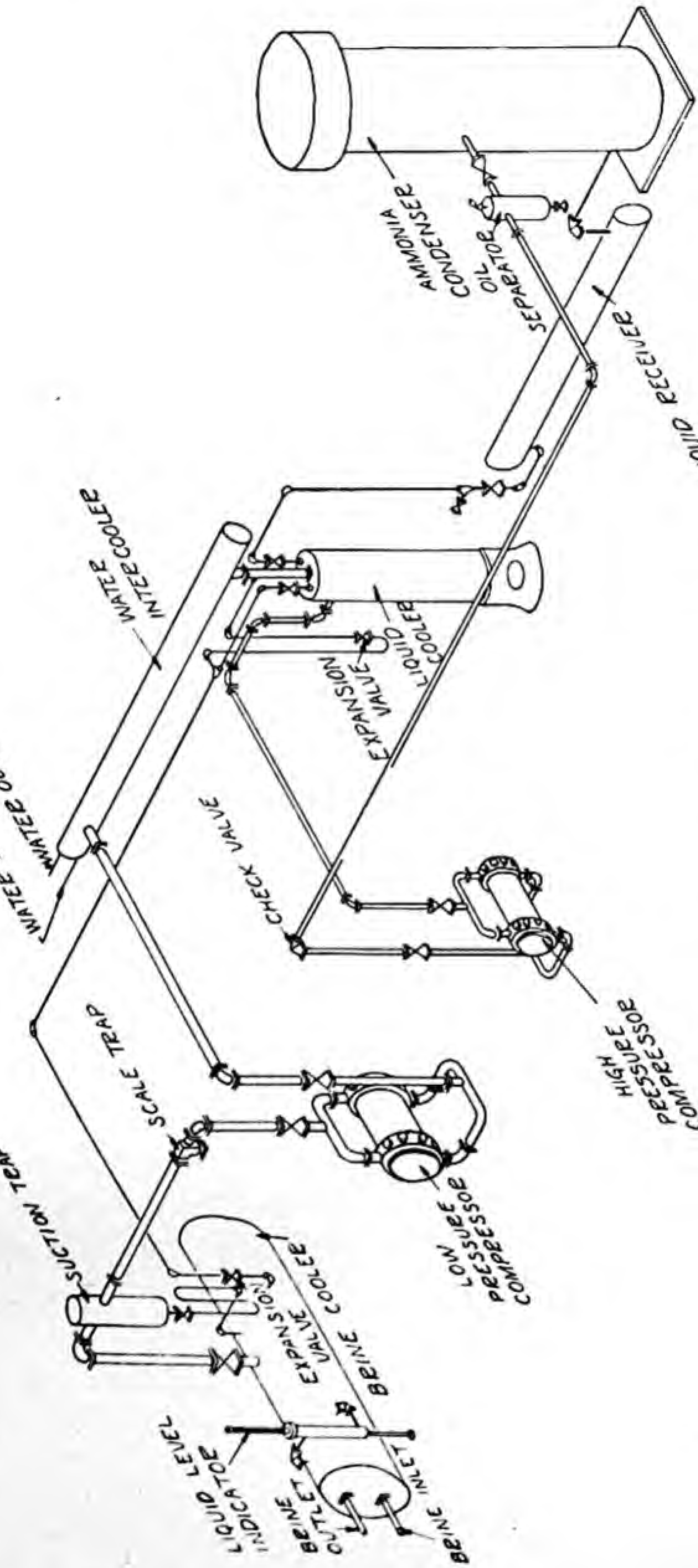


Fig. 38.—Vilter Multi-Stage Compression System.

flows by gravity into the so-called liquid cooler. It will be noticed that the compressed gas from the low pressure cylinder, cooled to as low a temperature as is possible by the water-cooled intercooler, mixes in direct contact with the liquid ammonia from the receiver, which is now at the pressure maintained in the intercooler which may be approximately 25 lbs. or 50 lbs. gauge, depending on the requirements. The saturation temperature in the trap depends on the pressure, of course, and will be about 12° F. to 34° F., as the case may be. Therefore, the gas leaving the trap, which has suitable drying baffles, will be cooled down to its temperature of saturation and the suction to the high-pressure cylinder will be in the best condition for efficient compression.

It will be noted from what precedes that some of the cooling in the intercooler was done by the liquid ammonia, and it may appear at first that no advantage can be derived from such a device. However, to make a typical example, the volume of one pound of ammonia at 50 lbs. gauge and a temperature of 84° F. is 5.03 cu. ft., whereas at the temperature of saturation it is 4.45 cu. ft.; or the volume is increased 13.0 per cent by the superheat present. But to cool this gas causes the vaporization of some liquid ammonia and the amount is equal to the difference (30.5 Btu.) in the total heat divided by the latent heat of vaporization (541.5 Btu.) or 5.6 per cent, thereby saving in piston displacement and work done in the high pressure cylinder the amount of the difference or 7.4 per cent. This is a typical example which will be substantially correct for any case which is calculated.

Referring again to the diagram, it will seem that the ammonia passing to the "refrigerator" has to flow through a coil pipe in the liquid cooler. The liquid cooler is open to the suction of the high pressure cylinder and to the trap or intercooler, and is maintained at the temperature of boiling ammonia at this pressure by the ammonia which drains off from the trap. The result is that the temperature of the ammonia at the refrigeration expansion valve is precooled, in consequence of which some ammonia outside the coils in the cooler is evaporated.

Here again it may be said that no advantage can be obtained by precooling, because ammonia is evaporated in the process, but these are not all of the facts. In every refrigerating plant, before useful refrigeration work may be performed, it is necessary to cool the liquid ammonia to the temperature at which boiling takes place in the refrigerating coils. If the ammonia is cooled to 60° F. in the condenser and the temperature in the cooling coils is zero, then 11.7 per cent of the liquid will be used up and vaporized before any useful refrigeration is performed. The vapor caused by this vaporization will pass into and clog the coils if no accumulator is used, and will subsequently return to the compressor. The vaporization takes place, but in two

stages, and that which occurs in the liquid cooler has to be compressed only in the high pressure cylinder. The saving is self evident.

It is well-known that the use of clearance in ammonia compressors does not decrease the efficiency of operation, but that it is a necessity for high speed compressors in order to cushion properly the reciprocating parts. The re-expansion of the gas in the clearance volume, of course, decreases the volumetric efficiency, but this re-expansion returns the work of compression to this piston. The net result is that it is only necessary (where clearance decreases the effective suction stroke) to increase the speed. However, in low temperature work the re-expansion in the clearance becomes excessive for single-stage compression, and the result is that quite a large increase in piston displacement would have to be made if the results were to be obtained.

Finally, there is the question of disintegration of ammonia for extreme conditions. Although the information is not definite as regards the stability of this chemical, yet it seems reasonably clear that, for extreme conditions of pressure and temperature, ammonia will tend to break up into hydrogen and nitrogen. These so-called "perfect gases" collect in the condenser and increase the condenser pressure and have to be purged out of the system as required. Usually a two-stage system keeps the operating temperature down to below 200° F.

Another type of two-stage compression system is illustrated in Fig. 39. Here the gas from the evaporator enters the low-pressure cylinder through the gas suction line and scale trap. After being compressed in the low-pressure cylinder the gas enters the flash gas receiver and intercooler. Liquid ammonia from the receiver is introduced into this vessel by means of a liquid level regulator. The liquid ammonia now under intercooler pressure is led to the evaporating unit. The flash gas receiver and intercooler then allows the flash gas liberated by cooling the liquid from receiver temperature to the flash gas receiver temperature to go to the high-pressure cylinder together with gas coming from the low-pressure cylinder. The flash receiver and intercooler also de-superheats the gas discharged by the low-pressure cylinder.

The high-pressure cylinder withdraws the gas from flash gas receiver and discharges it to the condenser.

Multiple Effect Compressors.—Harry Sloan gives the following method of figuring capacity and horsepower of multiple effect compressors:

Referring to Table No. 34 you will find that most of the details are given. Under the heading of "General Conditions" even temperatures have been selected for the two suction pressures and brings the suction pressure in decimal of pounds. The condenser pressure has been taken

at 180 lbs. and it has been assumed that the temperature of the liquid ammonia in the receiver would be 100° F. This is about 5° F. above the temperature of ammonia corresponding to 180 lbs. as given in the ammonia table. The Bureau of Standard Ammonia Tables were used, but no use was made of the Mollier Chart as it is probably easier to

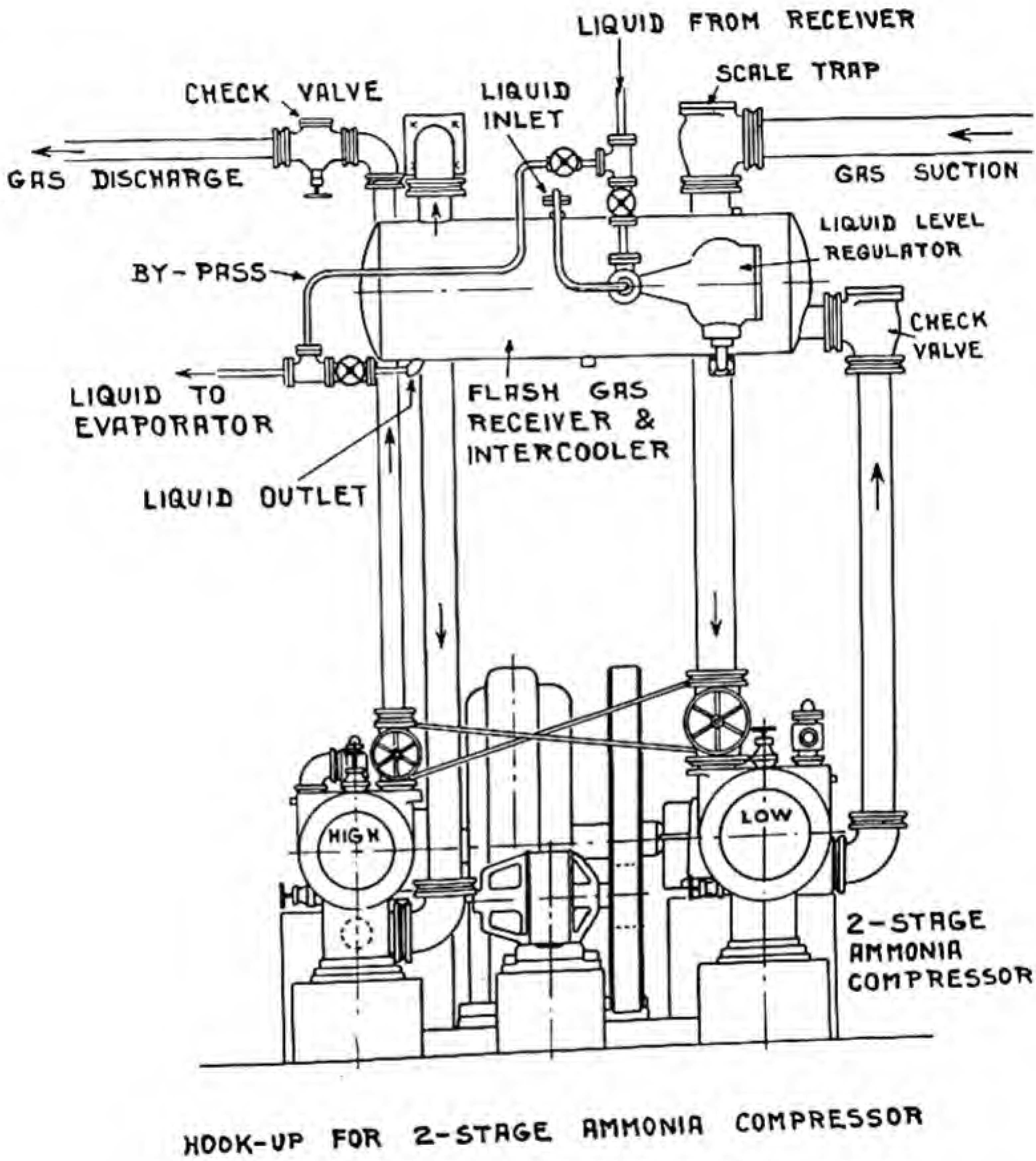


Fig. 39.—Vogt Two-Stage Ammonia Compressor.

follow a calculation such as this by taking the quantities from the tables rather than using and explaining the chart. There are very few assumptions made and quantities such as volumetric efficiency, etc., can be taken off from tables giving such data. When figuring the high-pressure displacement, it is necessary to take into consideration the rise in temperature of the gas when the low-pressure gas in the cylin-

der is compressed by the high-pressure gas. In the particular case 14.3 lbs. ammonia gas will be compressed to 37.9 lbs. by the high-pressure gas at 37.9 lbs. pressure. The final temperature under these conditions would be by calculation 49° F.

You will note that the low-pressure displacement calculated is in cubic feet per ton per minute, so all that is necessary to do is to multiply this value by the number of tons of low temperature work required.

TABLE 34.—MULTIPLE EFFECT COMPRESSOR AND RECEIVER

GENERAL CONDITIONS:	
Temperature at high suction pressure	24° F.
Temperature at low suction pressure	—2° F.
Pressure of compressor discharge, gauge	180.0 lbs.
Pressure of high suction vapor, gauge	37.9 lbs.
Pressure of low suction vapor, gauge	14.3
Port loss in displacement, per cent	2.5
Clearance, per cent	1.0
Temperature of liquid at low pressure expansion, 1° F. loss	25° F.
LOW PRESSURE DISPLACEMENT:	
Latent heat of evaporation of ammonia at —2° F. Btu. per lb.	570.4
Btu. to cool 1 lb. liquid from 25° to —2° F	29.7
Btu. per lb. available for useful refrigeration	540.7
Pounds ammonia per min. per ton refrigeration 200 ÷ 540.7	
Cu. ft. vapor per lb. at —2° F. (Ammonia Table)	9.541
Cu. ft. vapor ton per min. low pressure (9.541 × .37)	3.56
Volumetric efficiency of compressor	.79
Low pressure, compressor disp. cu. ft. per ton per min. allowing port loss	4.66
HIGH PRESSURE DISPLACEMENT:	
Available high pressure displacement cu. ft. per ton per min. with clearance added	4.71
Cu. ft. vapor per lb. at —2° F	9.541
Cu. ft. vapor per lb. at 24° F	5.443
Then the low pressure vapor would be compressed by the high pressure vapor $\frac{5.443}{9.541}$ = 57% or 43% of displacement available for hp.	
Cu. ft. available per ton per min. of high pressure vapor 4.71 × .43	2.025
Cu. ft. available allowing for heat of compression	1.94
Cu. ft. vapor required per ton per min. at 37.9 lbs. and 180 lbs. pressure	2.36
Tons refrigeration at high suction per ton at low suction $\frac{1.94}{2.36}$	.82
Tons refrigeration at high suction to cool liquid from 100° to 24° F.	.16
	.66
Tons refrigeration at high suction allowing 3% for ports and wire drawing	.64

This would give you the total displacement of your compressor per minute in cubic feet and then the size of cylinder can be determined. The calculation for the high-pressure capacity gives the result in tons of refrigeration per ton of low suction work, so under the particular conditions given in this table, if your calculation calls for 100 tons of low-pressure work, you will have 100 by .64 or 64 tons of high-pressure work making a total refrigeration of 164 tons. That is 100 tons at 14.3 lbs. suction and 64 tons at 37.9 lbs. suction pressure.

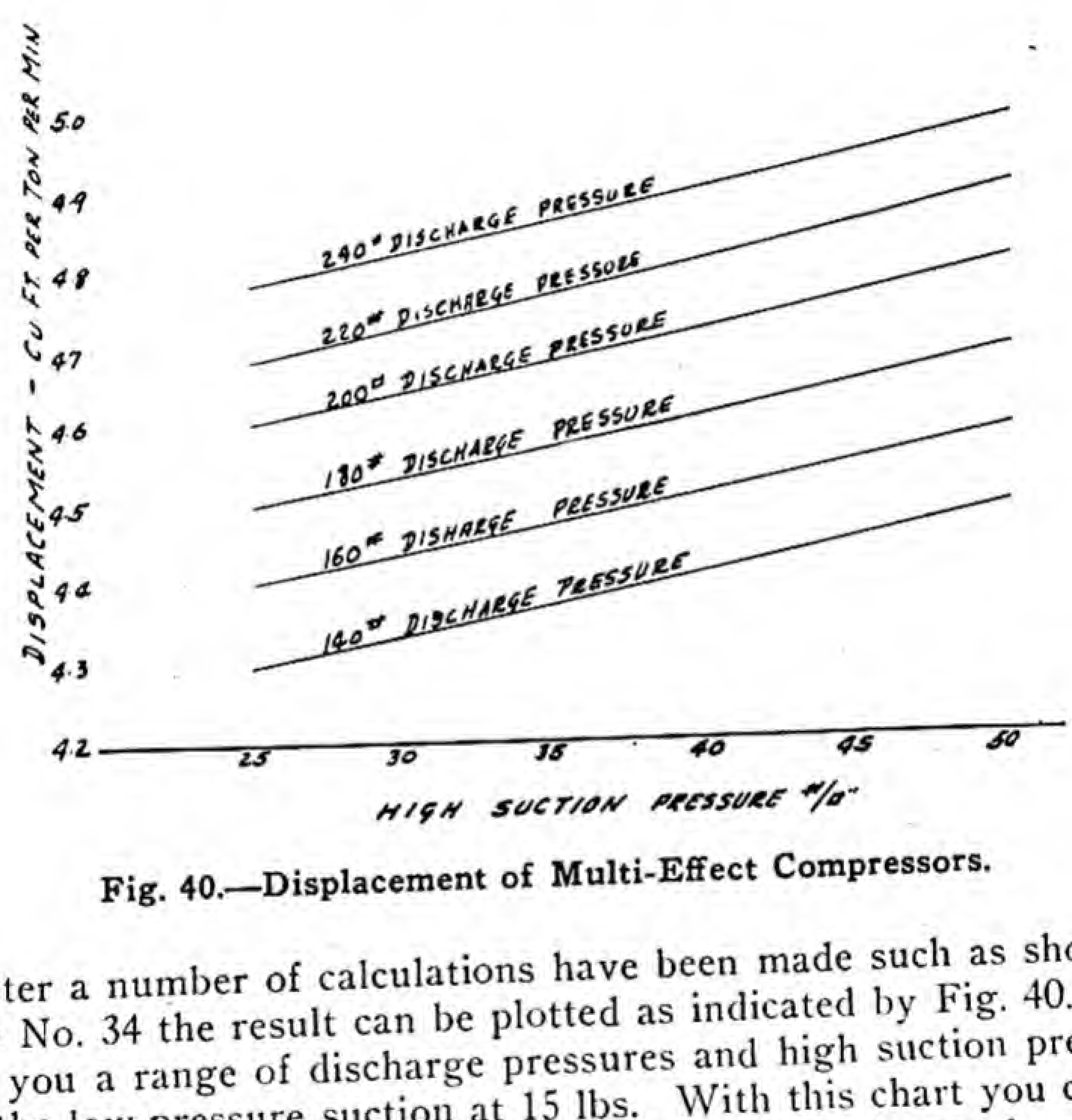


Fig. 40.—Displacement of Multi-Effect Compressors.

After a number of calculations have been made such as shown in Table No. 34 the result can be plotted as indicated by Fig. 40. This gives you a range of discharge pressures and high suction pressures with the low-pressure suction at 15 lbs. With this chart you can obtain the displacement required by the compressor in cu. ft. per ton per min.

The chart shown in Fig. 41 gives the additional tonnage which can be added to the tonnage of the low-pressure suction obtained in Fig. 40, and you will note that the condenser pressures and high suction pressures are of the same range as Fig. 40.

By the use of such charts, the calculation of the multiple effect compressor becomes a very easy matter.

The calculations are rather long and as often is the case, after you have calculated a compressor, find that alternates or changed condi-

tions perhaps will raise your high-pressure suction and it would be necessary to make a new calculation if you wanted to know the capacity. With the chart, this can be obtained with little effort.

The multiple-effect compressor should only be used where there is considerable difference between the high and low suction pressure. Where these run close together, say 5 lbs. or a little more, there would be no particular advantage in the multiple effect.

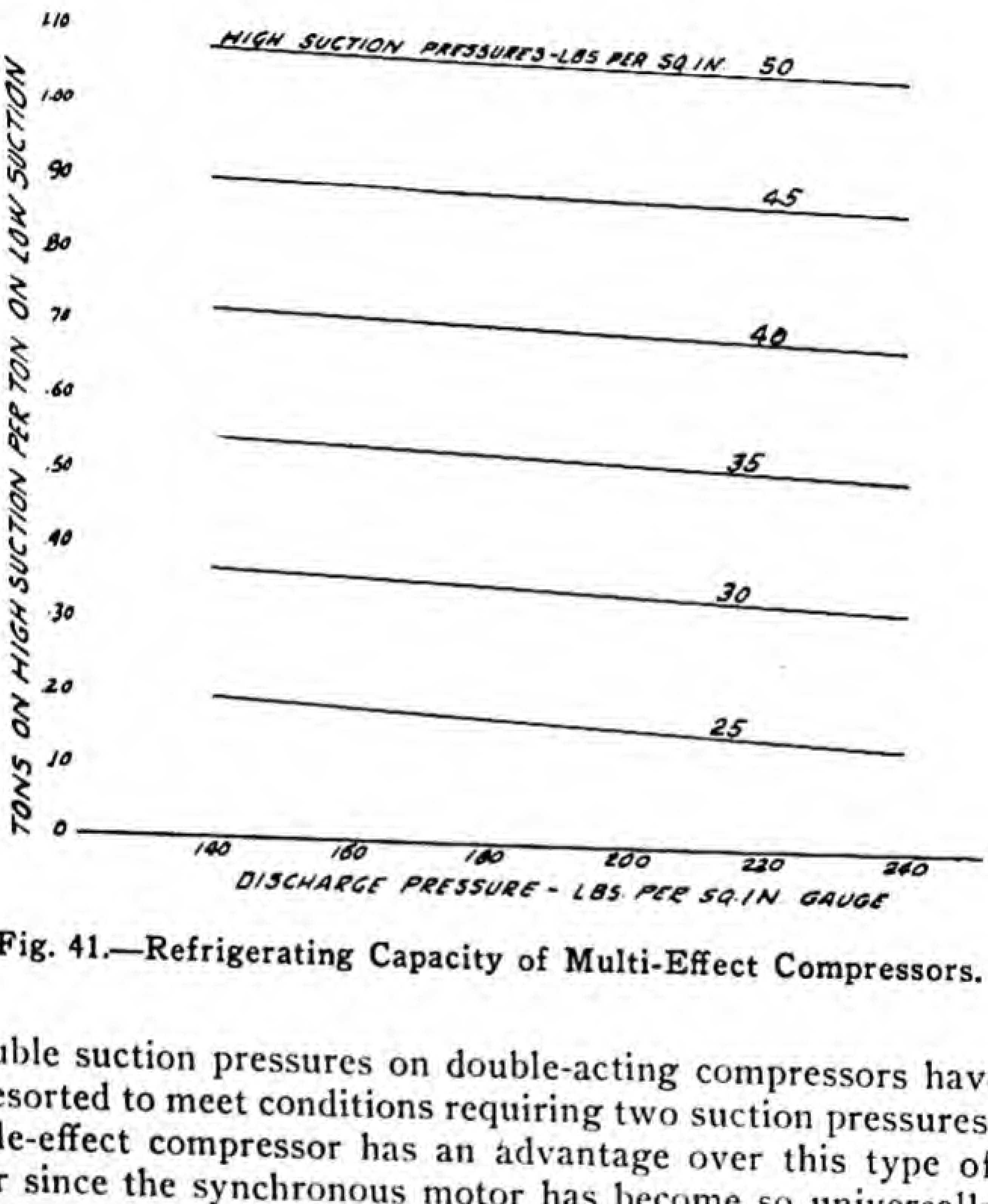


Fig. 41.—Refrigerating Capacity of Multi-Effect Compressors.

Double suction pressures on double-acting compressors have long been resorted to meet conditions requiring two suction pressures. The multiple-effect compressor has an advantage over this type of compressor since the synchronous motor has become so universally used for driving ammonia compressors. With the double suction on a compressor, the load is somewhat unbalanced and it will be necessary to incorporate more flywheel effect in the motor than would be necessary if the cylinders were made to operate on the multiple effect cycles. You can see that the work will be equal on each end of the cylinder and the work being perfectly balanced the motor will require less flywheel effect.

In regard to horsepower for multiple effect compressors, it was found that all kinds of statements and claims are made. We have found

that the capacity per cu. ft. of displacement is very materially increased, but from careful investigation the saving in horsepower is not so great. Dr. Inokuty, Tokyo, Japan, in a paper read before the Fifth International Congress of Refrigeration, found by careful tests, that, under the conditions he was operating with a CO_2 machine, for an increase in capacity of 49.7 per cent, the actual power by the indicator showed an increase of 35 per cent—this is a saving worth while as long as the plant is not complicated too much and is a comparatively large plant. The subject was also investigated by Prof. E. H. Lamb, and a paper

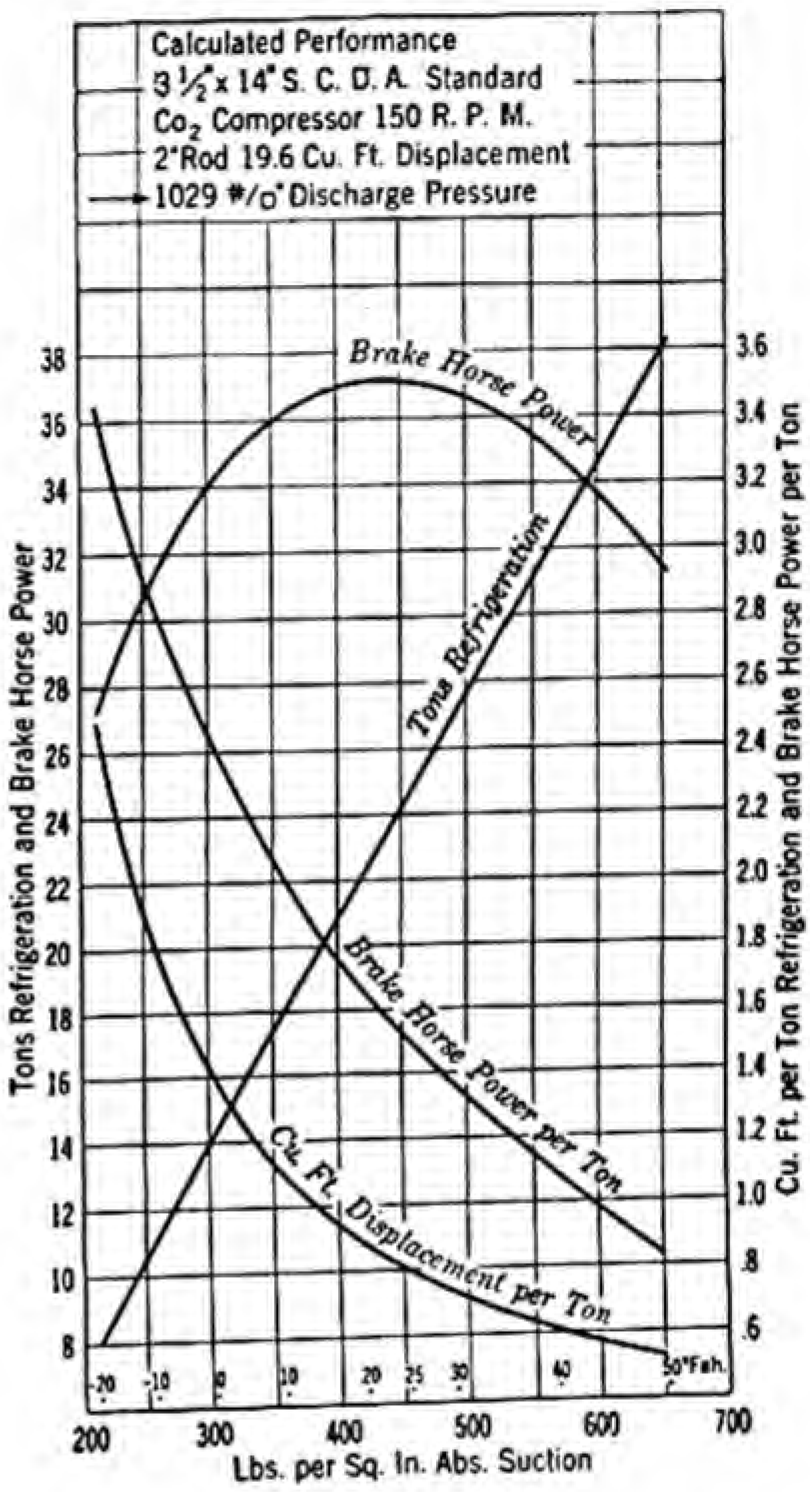


Fig. 42.

presented to the British Association, in 1926, giving a report on investigation of CO_2 compressors having different cycles. He claims that the saving in power is very small, but also found that the increase in capacity was very material under all conditions.

Voorhees, in his book, "Compression Refrigerating Machine," states that "the m.e.p. of a multiple-effect compressor indicator diagram is the m.e.p. of the high suction part thereof, plus the difference in high and

low suction pressure." This is a logical statement and undoubtedly, entirely correct. When a compressor is filled with a high-pressure gas, regardless of how the high-pressure was obtained, will require a horsepower corresponding to that pressure and would be similar to the ordinary compressor, except that instead of having a high suction pressure on the opposite side of the piston helping to compress the gas, we will

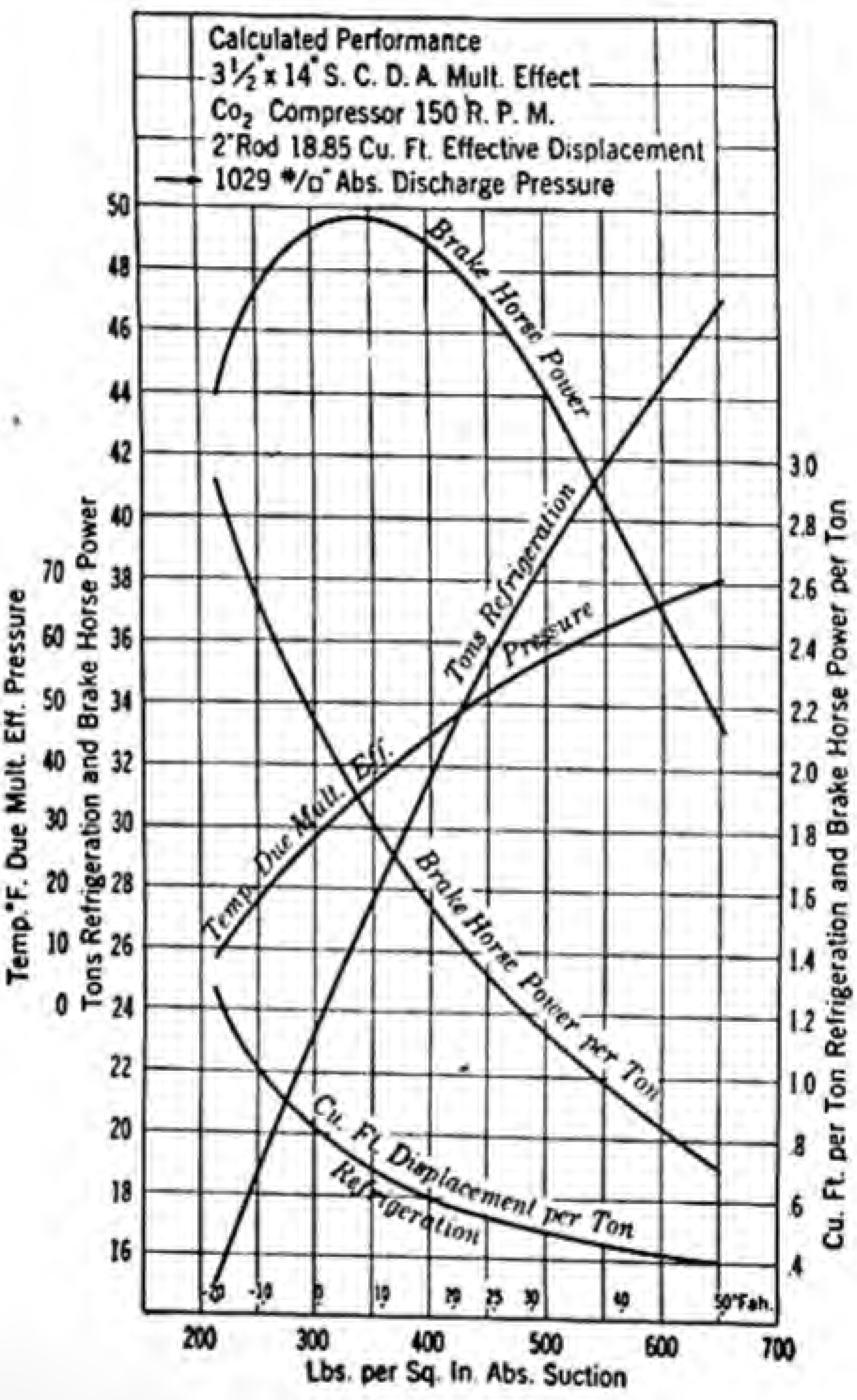


Fig. 43.

have the low suction pressure, so if we take the horsepower at the high suction pressure plus the horsepower loss, due to the lower suction pressure on the opposite side of the piston, we will have the correct horsepower for a multiple effect compressor.

Fig. 42 gives the calculated performance of a single cylinder horizontal, double-acting carbon dioxide compressor having a cylinder diameter of 3½ in. and a stroke of 14 in. This figure gives the total brake horsepower, brake horsepower per ton of refrigeration, cubic feet of displacement per ton refrigeration, and the total tons of re-

frigerating capacity, for various suction pressures, at a compressor speed of 150 r.p.m. and a condenser pressure of 1029 lbs. per sq. in.

Similar data are shown in Fig. 43 for a multiple-effect compressor of the same size and speed. This figure shows graphically the increase of refrigerating capacity and power requirements, due to the multiple suction pressure principle.

QUESTIONS ON CHAPTER IV.

1. Describe briefly the principles underlying the four phases of the cycle of operation of the compression system.

2. What is the function of the expansion valve and how does it accomplish this?

3. Describe in detail the condition of liquid ammonia just after passing through the expansion valve, if the saturated temperatures before and after the expansion valve are 95° and 0° F.

4. What would be the refrigerating effect of a pound of ammonia if the temperature of the liquid ammonia before and after the expansion valve is 80° F., and the temperature in the refrigerating coils is 5° F.?

5. How many pounds of ammonia must be evaporated per minute per ton of refrigeration if the absolute condenser pressure is 180 and the absolute evaporator pressure is 40 lbs.?

6. What theoretical displacement would be necessary in a cylinder of ammonia compressor operating between the pressures of 40 and 200 lbs. in the evaporator and condenser respectively? What displacement would be necessary if the evaporator pressure were reduced to 20 lbs. absolute?

7. What factors tend to lessen the amount of vapor that may get into a compressor cylinder?

8. Find the heat content of the superheated ammonia vapor and the temperature at the end of compression when the evaporator and condenser pressures are 30 and 155 lbs. per sq. in. abs., respectively, by the two methods of calculation.

9. By means of the values given in Tables 24 and 29, determine the actual refrigerating capacity and the indicated horsepower required of the ammonia compressor described in Example 2, Page 25, when the volumetric efficiency is 83 per cent and the suction and condenser pressures are 25 and 155 lbs. per sq. in. gauge respectively.

10. Calculate the amount of water required for cooling and condensing ammonia under conditions specified in Problem No. 8, if the water temperature increases from 55° to 68° F. in passing through the condenser.

CHAPTER V.

GENERAL PRINCIPLES OF THE ABSORPTION SYSTEM.

General Principle.—Broadly speaking, the general principle underlying the operation of the absorption refrigerating system is the same as that of the compression system, and may be stated as follows: The working substance is placed in such a condition that it will extract heat from the refrigerator and after this extraction of heat the working substance is placed in such a condition that it will give up the heat from the refrigerator and the heat added during the process, to a material at a temperature higher than that of the refrigerator space.

Attention has been previously directed to the similarity of the operation of the compression and absorption systems. In the compression system, the vapor from the evaporator in the refrigerator is compressed from low pressure to high pressure by mechanical means, while in the absorption system the pressure is increased by the application of heat to a liquid which contains the dissolved vapors from the evaporator.

Thus, water absorbs or dissolves the ammonia vapor from the evaporator at low temperature and pressure, and then it is made to give up or distill off the ammonia vapor at a higher temperature and pressure. This is the principle underlying the *absorption* part of the absorption system.

The cycle of operation of the elementary absorption system is as follows: The ammonia vapor from the evaporator is dissolved in a weak solution of ammonia and water, which is termed weak aqua ammonia. This absorption of the vapor takes place in the absorber, and since there is a change of state from vapor to liquid, the latent heats of condensation and absorption are liberated. This heat is taken out by water. Thus, the weak aqua becomes strong aqua, due to the absorption of ammonia from the evaporator. The strong aqua is then pumped from the absorber into a still or generator, in which, by the

application of heat, the ammonia is distilled into the condenser. The resulting weak aqua is piped back to the absorber to re-absorb ammonia again and thus complete this part of the cycle. The anhydrous liquid ammonia from the condsenser is fed through the expansion valve into the evaporator. By absorption of heat from the refrigerator, it is transformed into vapor again. It then returns to the absorber to be re-absorbed and then used over and over again.

From the above, it will be noted that the elementary absorption system has a condenser, expansion valve, and evaporator, similar to those of the compression system; and that the action in the absorber corresponds to the suction stroke of the mechanical compressor.

The action in the absorber and generator may be more readily understood by noting the characteristics of liquids containing dissolved vapors.

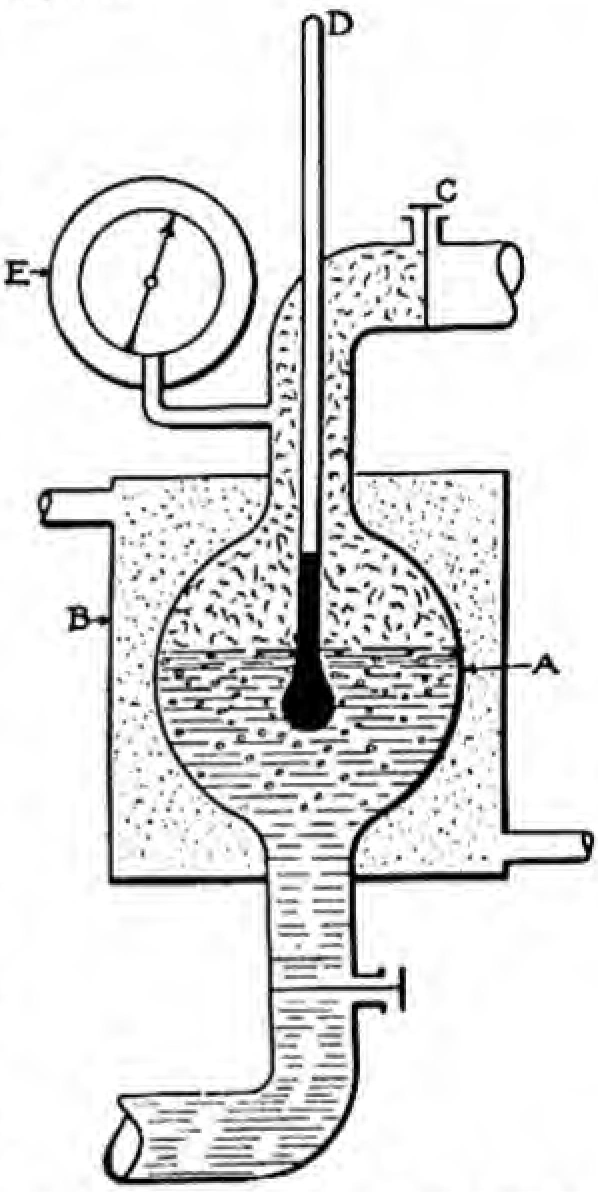


Fig. 44.—Aqua Ammonia Apparatus.

Aqua Ammonia.—All liquids will dissolve or absorb gases, but the extent of such a solution will depend upon the nature and condition of the substances. Generally, when there is a true solution of the gas in the liquid, the quantity of gas dissolved is increased by increasing the pressure or by lowering the temperature; also, the boiling temperature of the liquid is lowered by the absorption of the gas. Aqua ammonia is a solution of ammonia vapor in water, and is an example of this sort of solution.

The relationship of the temperatures, pressures, and concentrations of aqua ammonia have been investigated experimentally by Mollier. The results of these experiments have been translated and extended by Macintire. Table 18, Chapter III, shows the variation of temperature-pressure-concentration characteristics of aqua ammonia. The relation of the values represented in Table 18 may be expressed by the following formula:

$$(1) \qquad T_s = \frac{T_a}{0.00466X + 0.656}$$

in which T_s = the absolute temperature of the boiling solution

T_a = the absolute temperature of saturated ammonia corresponding to the pressure of the solution

X = per cent of ammonia by weight in the solution

0.00466 and 0.656 = constants.

Fig. 44 will be helpful in obtaining a physical conception of the variation of the properties of aqua ammonia. If the container *A* is filled with water as shown, and steam at 225° F. is turned into the heater *B*, the water will boil at a temperature of 212° F. as would be indicated by the thermometer *D*. The valve *C* in this case is open to the atmosphere, and the pressure gauge would read zero pounds. Now, if the water is removed and a solution of 20 per cent ammonia and 80 per cent water is put into the container *A*, leaving valve *C* open, the solution would boil at a temperature several degrees below 212° F. In this case, the thermometer *D* would indicate a boiling temperature of only 117.6° F. This temperature may be read from Table 18, or calculated by the above formula as follows:

$$T_a = 459.6 + (-27.3) = 432.3° \text{ abs.}$$
$$X = 20$$
$$\text{therefore } T_s = \frac{432.3}{0.00466 \times 20 + 0.656} = 577.2° \text{ abs.}$$
$$\text{and F. Temp.} = 577.2 - 459.6 = 117.6° \text{ F.}$$

The following tabulation will show how the temperature of the boiling point is lowered as the concentration is increased at a constant pressure of 15 lbs. per sq. in. abs.:

Abs. Pressure lbs. per sq. in.	Per cent Water	Per cent Ammonia	Boiling Temp. ° F.
15	100	0	212.0
15	90	10	155.0
15	80	20	117.6
15	70	30	83.8
15	60	40	53.5
15	50	50	26.9
15	0	100	−27.3

In a similar manner, the variation of the concentration with the increase of pressure may be noted. As in the previous example, pure water under a pressure of the atmosphere will boil at 212° F. The water is pure and contains no ammonia. Now, if the pressure is increased to 78 lbs. per sq. in. abs., and ammonia is added to the water, the solution would have to contain 20 per cent ammonia before it would boil at 212° F. The following tabulation shows how the increase of pressure increases the amount of ammonia that the solution could hold at a constant boiling temperature of 212° F.:

Boiling Temp. ° F.	Per cent Water	Per cent Ammonia	Abs. Pressure lbs. per sq. in.
212	100	0	15.
212	90	10	40
212	80	20	78
212	70	30	140
212	60	40	...
212	50	50	...
212	0	100	890

In a like manner, a solution of 24 per cent ammonia and 76 per cent water may be put into the container *A*, and steam of various temperatures may be used in the heater *B*. If a pressure of 20 lbs. per sq. in. abs. is imposed upon the solution, it will boil at 117° F., and if the pressure is increased to 200 lbs., it will boil at 265° F. This illustrates the physical law that states when the pressure of a liquid is increased, the temperature of the boiling or condensing point is increased in proportion, and if the pressure is decreased, the boiling or condensing temperature is lowered. The following tabulation shows the variation of the boiling temperature with the increase of pressure on the aqua ammonia when the concentration is 24 per cent:

Per cent Ammonia	Per cent Water	Abs. Pressure lbs. per sq. in.	Boiling Temp. ° F.
24	76	20	117
24	76	40	155
24	76	60	180
24	76	80	198
24	76	100	213
24	76	120	225
24	76	140	237
24	76	160	247
24	76	180	255
24	76	200	265

Table 41 showing the properties of saturated steam has been incorporated at the end of this chapter for reference purposes.

Specific Gravity of Aqua Ammonia.—The commercial and practical method for determining the strength of aqua ammonia solutions is by the use of the hydrometer. Most hydrometer scales are graduated in degrees Beaumé which is a confusing and arbitrary scale. On this Beaumé scale, the specific gravity corresponding to 1.00 is placed at 10.0.

The relation between the specific gravities of liquids lighter than water and the Beaumé readings is expressed by the following formula:

$$\text{Degrees Beaumé} = \frac{140}{\text{spec. grav.}} - 130 \qquad (2)$$

It is evident that since the specific gravity readings are the ones that are important, the use of the Beaumé hydrometer and the subsequent conversions to true specific gravities present more chances for errors. Also, it is evident that the Beaumé hydrometer should only be used for making practical tests and that more refined methods of determination should be pursued in making scientific tests.

In addition to the specific gravity of aqua ammonia solutions, it is further evident that the percentages of ammonia and water in the solutions are important. The percentage of ammonia in the solution will

vary with the specific gravity, increasing as the specific gravity decreases.

The relation between the specific gravity and concentration of a solution of ammonia and water is expressed approximately by the following formula:

$$\text{(3)} \qquad \text{Spec. Grav.} = 1 - \frac{4.3}{1000}\left[X - \frac{X^2}{100} + \frac{X^3}{10{,}000}\right]$$

Where X = concentration in per cent.

TABLE 35.—SPECIFIC GRAVITY OF AQUA AMMONIA.

By W. C. FERGUSON

Degrees Beaumé	Specific Gravity	Per cent Ammonia	Degrees Beaumé	Specific Gravity	Per cent Ammonia
10.00	1.0000	0.00	19.50	0.9365	16.80
10.25	0.9982	0.40	19.75	0.9349	17.28
10.50	0.9964	0.80	20.00	0.9333	17.76
10.75	0.9947	1.21	20.25	0.9318	18.24
11.00	0.9929	1.62	20.50	0.9302	18.72
11.25	0.9912	2.04	20.75	0.9287	19.20
11.50	0.9894	2.46	21.00	0.9272	19.68
11.75	0.9876	2.88	21.25	0.9256	20.16
12.00	0.9859	3.30	21.50	0.9241	20.64
12.25	0.9842	3.73	21.75	0.9226	21.12
12.50	0.9825	4.16	22.00	0.9211	21.60
12.75	0.9807	4.59	22.25	0.9195	22.08
13.00	0.9790	5.02	22.50	0.9180	22.56
13.25	0.9773	5.45	22.75	0.9165	23.04
13.50	0.9756	5.88	23.00	0.9150	23.52
13.75	0.9739	6.31	23.25	0.9135	24.01
14.00	0.9722	6.74	23.50	0.9121	24.50
14.25	0.9705	7.17	23.75	0.9106	24.99
14.50	0.9689	7.61	24.00	0.9091	25.48
14.75	0.9672	8.05	24.25	0.9076	25.97
15.00	0.9655	8.49	24.50	0.9061	26.46
15.25	0.9639	8.93	24.75	0.9047	26.95
15.50	0.9622	9.38	25.00	0.9032	27.44
15.75	0.9605	9.83	25.25	0.9018	27.93
16.00	0.9589	10.28	25.50	0.9003	28.42
			25.75	0.8989	28.91
16.25	0.9573	10.73	26.00	0.8974	29.40
16.50	0.9556	11.18	26.25	0.8960	29.89
16.75	0.9540	11.64	26.50	0.8946	30.38
17.00	0.9524	12.10	26.75	0.8931	30.87
17.25	0.9508	12.56	27.00	0.8917	31.36
17.50	0.9492	13.02	27.25	0.8903	31.85
17.75	0.9475	13.49	27.50	0.8889	32.34
18.00	0.9459	13.96	27.75	0.8875	32.83
18.25	0.9444	14.43	28.00	0.8861	33.32
18.50	0.9428	14.90	28.25	0.8847	33.81
18.75	0.9412	15.37	28.50	0.8833	34.30
19.00	0.9396	15.84	28.75	0.8819	34.79
19.25	0.9380	16.32	29.00	0.8805	35.28

The relation between the specific gravity degrees Beaumé and concentrations of aqua ammonia is shown by Tables 35 and 36, Table 35 showing even degrees Beaumé and Table 36 showing even per cents of ammonia. The per cents given in these tables are percentage by

weight of ammonia in the solution. The specific gravity determinations in these tables were made at 60° F. and compared with water at 60° F. Since specific gravities may be determined at temperatures other than 60° F. and since the specific gravities would be affected by the change of volume due to the change of temperature, certain corrections must be allowed. Thus, Table 37 indicates corrections which must be made for each degree of temperature below 60° F. and for each degree of temperature above 60° F.

TABLE 36.—SPECIFIC GRAVITY OF AQUA AMMONIA.

Per cent Ammonia	Specific Gravity	Degree Beaumé	Per cent Ammonia	Specific Gravity	Degree Beaumé
0	1.000	10.0	20	0.925	21.7
2	0.986	12.0	22	0.919	22.8
4	0.979	13.0	24	0.913	23.9
6	0.972	14.0	26	0.907	24.9
8	0.966	15.0	28	0.902	25.7
10	0.960	16.0	30	0.897	26.6
12	0.953	17.1	32	0.892	27.5
14	0.945	18.3	34	0.888	28.4
16	0.938	19.5	36	0.884	29.3
18	0.931	20.7	38	0.880	30.2

Bulletin 146, of University of Illinois Experiment Station, gives the following in reference to the pressures of aqua ammonia solutions.

Because of the complexity of the equations which have been formulated as expressions for the calculation of total and partial pressures of the ammonia-water system, and also with the aim of making the work of more practical value, tables containing the necessary data have been appended.

TABLE 37.—ALLOWANCES FOR TEMPERATURES OF AQUA AMMONIA SOLUTIONS.

Degrees Beaumé	Corrections to be added when temp. is below 60° F.		Corrections to be subtracted when temp. is above 60° F.		
	40° F.	50° F.	70° F.	80 F.	90° F.
14°	0.015° Be.	0.017° Be.	0.020° Be.	0.022° Be.	0.024° Be.
16°	0.021° Be.	0.023° Be.	0.026° Be.	0.028° Be.	0.030° Be.
18°	0.027° Be.	0.029° Be.	0.031° Be.	0.033° Be.	0.035° Be.
20°	0.033° Be.	0.036° Be.	0.037° Be.	0.038° Be.	0.040° Be.
22°	0.039° Be.	0.042° Be.	0.043° Be.	0.045° Be.	0.047° Be.
26°	0.053° Be.	0.057° Be.	0.057° Be.	0.059° Be.	....

In Table 38 are recorded the total pressure of aqua ammonia over the entire range of concentration in 5 per cent molal increments between the temperature limits of 32° F. and 250° F. in 10 degree increments, relying on the experimental data of the Bureau of Standards for anhydrous ammonia, by the use of Equation (1), Chapter II, Bull. 146. The pressures shown are in pounds per sq. in. abs. The results calculated from the equation have been checked by the method of suc-

TABLE 38.—TOTAL VAPOR PRESSURES OF AQUA AMMONIA.

Pressures are in Pounds per Square Inch Absolute.

Temp. Deg. F.	Molal Concentration of Ammonia in the Solutions in Percentages										
	0	5	10	15	20	25	30	35	40	45	50
32	0.09	0.34	0.60	0.97	1.58	2.60	4.20	6.54	9.93	14.18	19.40
40	0.12	0.45	0.77	1.24	2.01	3.25	5.21	8.06	12.05	17.20	23.39
50	0.18	0.64	1.05	1.65	2.67	4.29	6.75	10.35	15.34	21.65	29.26
60	0.26	0.86	1.42	2.21	3.51	5.55	8.65	13.22	19.30	27.05	36.26
70	0.36	1.17	1.84	2.90	4.56	7.13	11.01	16.56	24.05	33.39	44.42
80	0.51	1.52	2.43	3.76	5.85	9.06	13.86	20.61	29.69	40.96	54.08
90	0.70	2.02	3.15	4.83	7.43	11.40	17.23	25.48	36.34	49.82	65.32
100	0.95	2.62	4.05	6.13	9.34	14.22	21.32	31.16	44.12	59.99	78.30
110	1.27	3.34	5.14	7.72	11.64	17.58	26.07	37.81	53.16	71.87	93.19
120	1.69	4.27	6.46	9.63	14.42	21.54	31.69	45.62	63.59	85.33	110.20
130	2.22	5.38	8.07	11.91	17.67	26.20	38.25	54.55	75.55	100.86	129.50
140	2.89	6.70	9.98	14.63	21.49	31.54	45.73	64.78	89.19	118.24	151.30
150	3.72	8.29	12.23	17.81	26.00	37.81	54.43	76.61	104.65	138.10	175.40
160	4.74	10.16	14.92	21.54	31.16	45.02	64.25	89.88	122.10	160.20	202.70
170	5.99	12.41	18.01	25.87	37.11	53.27	75.55	104.84	141.75	185.10	233.20
180	7.51	15.00	21.65	30.86	44.02	62.68	88.17	121.68	163.70	212.60	267.00
190	9.34	18.06	25.87	36.60	51.81	73.32	102.56	140.75	188.10	243.30	304.30
200	11.53	21.60	30.72	43.14	60.62	85.33	118.68	161.81	215.20	277.00	345.50
210	14.12	25.61	36.26	50.58	70.72	98.80	136.42	185.10	245.10	314.50	390.70
220	17.19	30.27	42.47	59.00	81.91	113.81	156.41	211.24	278.20	355.10	439.60
230	20.78	35.59	49.60	68.46	94.43	130.64	178.28	239.70	314.50	400.20	493.40
240	24.97	41.52	57.65	78.91	108.60	149.20	202.74	270.92	354.10	448.90	552.30
250	29.83	48.32	66.67	90.74	124.08	169.48	229.62	305.60	397.60	502.40	

TABLE 38.—TOTAL VAPOR PRESSURES OF AQUA AMMONIA.—(Concluded.)

Pressures are in Pounds per Square Inch Absolute.

Temp. Deg. F.	Molal Concentration of Ammonia in the Solutions in Percentages									
	55	60	65	70	75	80	85	90	95	100
32	25.16	31.16	36.77	42.72	45.94	49.28	52.14	54.90	58.01	62.29
40	30.20	37.20	43.73	49.60	54.43	58.33	61.64	64.78	68.32	73.32
50	37.54	45.93	53.85	60.87	66.67	71.29	75.25	79.07	83.41	89.19
60	46.23	56.32	65.90	74.06	80.96	86.49	91.08	95.69	100.66	107.60
70	56.44	68.46	79.54	89.36	97.51	104.08	109.60	114.86	120.63	128.80
80	68.19	82.55	95.69	107.20	116.54	124.30	130.64	136.40	143.72	153.00
90	81.91	98.61	114.02	127.42	138.34	147.15	154.56	161.81	169.76	180.60
100	97.68	117.17	135.01	150.50	163.16	173.40	182.10	190.22	199.22	211.90
110	115.70	138.10	158.84	176.54	191.15	203.26	212.89	222.34	232.85	247.00
120	136.20	162.08	185.70	206.29	222.68	236.37	247.38	258.40	270.10	286.40
130	159.40	189.00	215.88	239.33	258.40	273.30	286.40	298.67	311.90	330.30
140	185.40	219.28	249.66	276.15	297.81	315.00	329.40	343.20	358.60	379.10
150	214.50	252.65	287.24	317.30	341.70	361.10	377.10	392.80	409.80	433.20
160	247.00	290.18	329.40	363.10	390.20	412.20	430.40	447.80	466.60	492.80
170	283.10	331.70	375.60	413.30	443.70	467.80	488.70	508.20	528.80	558.40
180	323.10	377.10	426.60	468.40	502.40	529.50	552.30	......	......	......
190	367.10	427.70	482.50	528.80	......	......	......	......	......	......
200	415.10	483.00	543.60	......	......	......	......	......	......	......
210	468.40	542.90	......	......	......	......	......	......	......	......
220	525.50	......	......	......	......	......	......	......	......	......
230	......	......	......	......	......	......	......	......	......	......
240	......	......	......	......	......	......	......	......	......	......
250	......	......	......	......	......	......	......	......	......	......

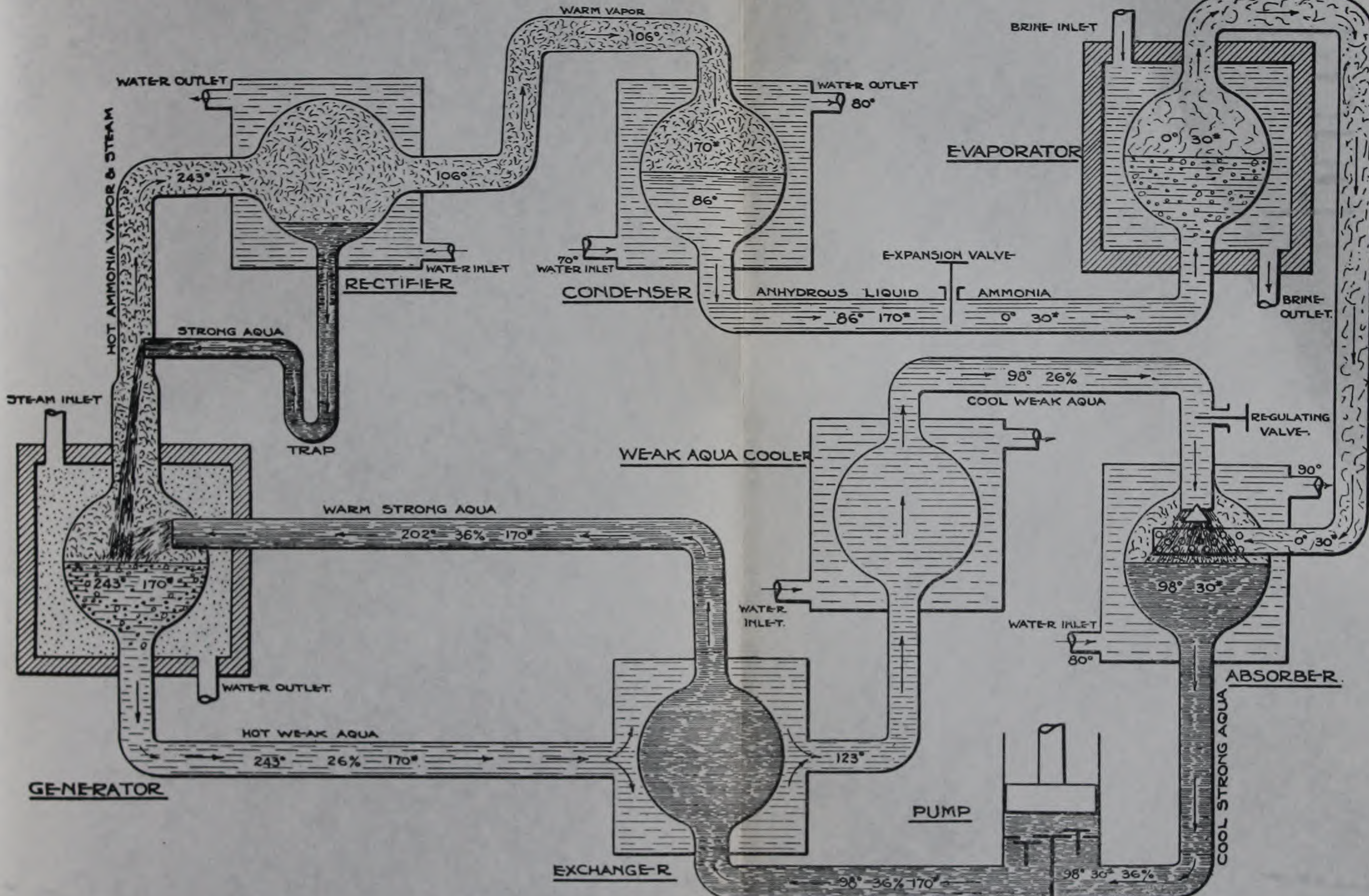


Fig. 45.—Diagram of Absorption Refrigerating Machine.

TABLE 39.—PARTIAL PRESSURES OF WATER VAPOR ABOVE AQUA AMMONIA.
Pressures are in Pounds per Square Inch Absolute.

Temp. Deg. F.	Molal Concentration of Ammonia in the Solutions in Percentages										
	0	5	10	15	20	25	30	35	40	45	50
32	0.09	0.084	0.079	0.074	0.070	0.065	0.060	0.056	0.051	0.047	0.042
40	0.12	0.115	0.108	0.101	0.095	0.089	0.083	0.076	0.070	0.064	0.058
50	0.18	0.17	0.16	0.15	0.14	0.13	0.12	0.11	0.10	0.094	0.085
60	0.26	0.24	0.23	0.21	0.20	0.19	0.17	0.16	0.15	0.13	0.12
70	0.36	0.34	0.32	0.30	0.28	0.26	0.25	0.23	0.21	0.19	0.17
80	0.51	0.48	0.45	0.42	0.40	0.37	0.34	0.32	0.29	0.27	0.24
90	0.70	0.66	0.63	0.58	0.55	0.51	0.47	0.44	0.40	0.37	0.33
100	0.95	0.90	0.85	0.79	0.74	0.69	0.64	0.59	0.55	0.50	0.45
110	1.27	1.20	1.14	1.07	1.00	0.93	0.86	0.80	0.73	0.67	0.60
120	1.69	1.60	1.51	1.42	1.33	1.24	1.15	1.06	0.97	0.89	0.80
130	2.22	2.10	1.98	1.86	1.74	1.62	1.51	1.39	1.28	1.17	1.05
140	2.89	2.73	2.57	2.42	2.26	2.11	1.96	1.81	1.66	1.52	1.37
150	3.72	3.51	3.31	3.11	2.91	2.72	2.52	2.33	2.14	1.95	1.76
160	4.74	4.48	4.22	3.97	3.71	3.46	3.22	2.97	2.73	2.49	2.25
170	5.99	5.66	5.34	5.02	4.70	4.38	4.07	3.75	3.45	3.15	2.84
180	7.51	7.10	6.69	6.30	5.89	5.49	5.10	4.71	4.33	3.94	3.57
190	9.34	8.83	8.32	7.82	7.32	6.83	6.34	5.86	5.38	4.91	4.44
200	11.53	10.90	10.27	9.65	9.04	8.43	7.83	7.23	6.64	6.06	5.48
210	14.12	13.35	12.58	11.82	11.07	10.32	9.59	8.86	8.13	7.42	6.71
220	17.19	16.25	15.32	14.39	13.48	12.57	11.67	10.78	9.90	9.03	8.17
230	20.78	19.64	18.51	17.40	16.29	15.19	14.11	13.03	11.97	10.91	9.87
240	24.97	23.60	22.25	20.91	19.58	18.26	16.95	15.66	14.38	13.12	11.86
250	29.83	28.20	26.58	25.00	23.39	21.82	20.25	18.71	17.18	15.67	.....

TABLE 39.—PARTIAL PRESSURES OF WATER VAPOR ABOVE AQUA AMMONIA.—(Concluded.)

Pressures are in Pounds per Square Inch Absolute.

Temp. Deg. F.	Molal Concentration of Ammonia in the Solutions in Percentages									
	55	60	65	70	75	80	85	90	95	100
32	0.038	0.034	0.030	0.025	0.021	0.017	0.013	0.008	0.004	0.00
40	0.052	0.046	0.040	0.035	0.029	0.023	0.015	0.012	0.006	....
50	0.076	0.068	0.059	0.051	0.042	0.034	0.025	0.017	0.008	....
60	0.11	0.097	0.085	0.073	0.061	0.049	0.037	0.024	0.012	....
70	0.15	0.14	0.12	0.10	0.086	0.069	0.052	0.034	0.017	....
80	0.22	0.19	0.17	0.14	0.12	0.096	0.072	0.048	0.024	....
90	0.30	0.26	0.23	0.20	0.16	0.13	0.10	0.066	0.033	....
100	0.41	0.36	0.31	0.27	0.22	0.18	0.13	0.090	0.045	....
110	0.54	0.48	0.42	0.36	0.30	0.24	0.18	0.120	0.061	....
120	0.72	0.64	0.56	0.48	0.40	0.32	0.24	0.160	0.081	....
130	0.95	0.84	0.74	0.63	0.53	0.42	0.32	0.210	0.100	....
140	1.23	1.10	0.96	0.82	0.69	0.55	0.41	0.270	0.140	....
150	1.59	1.41	1.24	1.06	0.88	0.71	0.53	0.350	0.180	....
160	2.02	1.80	1.58	1.35	1.12	0.90	0.67	0.450	0.220	....
170	2.56	2.28	1.99	1.71	1.42	1.13	0.85	0.570	0.300	....
180	3.21	2.85	2.50	2.14	1.77	1.42	1.06	.....	.....	....
190	3.99	3.55	3.10	2.65	.....	.....	.....	.....	.....	....
200	4.93	4.38	3.81	.....	.....	.....	.....	.....	.....	....
210	6.04	5.34	.....	.....	.....	.....	.....	.....	.....	....
220	7.31	.....	.....	.....	.....	.....	.....	.....	.....	....
230	.....	.....	.....	.....	.....	.....	.....	.....	.....	....
240	.....	.....	.....	.....	.....	.....	.....	.....	.....	....
250	.....	.....	.....	.....	.....	.....	.....	.....	.....	....

TABLE 40.—PARTIAL PRESSURES OF AMMONIA ABOVE AQUA AMMONIA

Pressures are in Pounds per Square Inch Absolute.

Temp. Deg. F.	Molal Concentration of Ammonia in the Solutions in Percentages									
	5	10	15	20	25	30	35	40	45	50
32	0.11	0.40	0.90	1.51	2.54	4.14	6.48	8.88	14.13	19.36
40	0.22	0.58	1.14	1.92	3.16	5.13	7.98	11.98	17.14	23.33
50	0.47	0.89	1.51	2.53	4.16	6.63	10.24	15.24	21.56	29.18
60	0.62	1.19	2.00	3.31	5.36	8.48	13.06	19.15	26.92	36.14
70	0.83	1.32	2.69	4.28	6.67	10.76	16.33	23.84	33.20	44.25
80	1.06	1.98	3.34	5.45	8.69	13.52	20.29	29.40	40.69	53.84
90	1.36	2.52	4.25	6.88	10.89	16.75	25.04	35.94	49.45	64.99
100	1.72	3.20	5.34	8.60	13.53	20.68	30.57	43.57	59.49	77.85
110	2.14	4.00	6.65	10.64	16.65	25.21	37.01	52.43	71.20	92.59
120	2.67	4.95	8.21	13.09	20.20	30.54	44.56	62.62	84.44	109.40
130	3.28	6.09	10.05	15.93	24.58	36.74	53.16	74.27	99.69	128.45
140	3.98	7.41	12.21	19.23	29.43	43.77	62.97	87.53	116.72	149.93
150	4.78	8.92	14.70	23.09	35.09	51.91	74.28	102.51	136.15	173.64
160	5.68	10.70	17.57	27.45	41.56	61.03	86.91	119.37	157.71	200.45
170	6.75	12.72	20.85	32.41	48.89	71.48	101.09	138.30	181.95	230.36
180	7.90	14.96	24.56	38.13	57.19	83.07	116.97	159.37	208.66	263.43
190	9.23	17.55	28.78	44.49	66.39	96.22	134.89	182.72	238.39	299.86
200	10.70	20.44	33.39	51.58	76.90	110.85	154.58	208.56	270.94	340.02
210	12.26	23.68	38.76	59.65	88.48	126.83	176.24	236.97	307.08	383.99
220	14.02	27.15	44.61	68.43	101.24	144.74	200.46	268.30	346.07	431.43
230	15.95	31.09	51.06	78.14	115.45	164.17	226.67	302.50	389.29	483.53
240	17.92	35.40	58.00	89.02	130.94	185.79	255.26	339.72	435.78	430.44
250	20.13	40.09	65.74	100.69	147.66	209.37	286.89	380.42	486.73	......

TABLE 40.—PARTIAL PRESSURES OF AMMONIA ABOVE AQUA AMMONIA.—(Concluded.)
Pressures are in Pounds per Square Inch Absolute.

Temp. Deg. F.	Molal Concentration of Ammonia in the Solutions in Percentages									
	55	60	65	70	75	80	85	90	95	100
32	25.12	31.13	36.74	42.70	45.92	49.26	52.13	54.89	58.01	0.00
40	30.15	37.15	43.69	49.57	54.40	58.31	61.62	64.77	68.31	....
50	37.46	45.85	53.79	60.82	66.63	71.26	75.23	79.05	83.40	....
60	46.12	56.22	65.82	73.99	80.90	86.44	91.04	95.67	100.65	....
70	56.39	68.32	79.42	89.26	97.42	104.01	109.55	114.83	120.61	....
80	67.97	82.36	95.52	107.06	116.42	124.20	130.57	136.85	143.70	....
90	81.61	98.35	113.79	127.22	138.18	147.02	154.46	161.74	169.73	....
100	97.27	116.81	134.70	150.23	162.94	173.22	181.97	190.13	199.18	....
110	115.16	137.62	158.42	176.18	190.85	203.02	212.71	222.22	232.79	....
120	135.48	161.44	185.14	205.81	222.28	236.05	247.14	258.24	270.02	....
130	158.45	188.16	215.14	238.70	257.87	272.88	286.08	298.46	311.80	....
140	184.17	218.18	248.70	275.30	297.14	314.45	328.99	342.93	358.44	....
150	212.91	251.24	285.90	316.24	340.82	360.39	376.57	392.45	409.62	....
160	244.98	288.38	327.82	361.75	389.08	411.30	430.73	447.35	466.38	....
170	280.54	329.42	373.61	411.60	442.28	466.67	487.85	507.64	528.72	....
180	319.89	374.25	424.10	466.26	500.63	528.08	551.24	......	......	....
190	363.11	424.15	479.40	526.15	......	......	......	......	......	....
200	410.17	478.62	539.79	......	......	......	......	......	......	....
210	462.36	537.56	......	......	......	......	......	......	......	....
220	518.19	......	......	......	......	......	......	......	......	....
230	......	......	......	......	......	......	......	......	......	....
240	......	......	......	......	......	......	......	......	......	....
250	......	......	......	......	......	......	......	......	......	....

cessive differences. These values may be readily applied to solutions when the weight per cent concentration of ammonia is known by means of the conversion factors given in Table 38.

Table 39 contains the corresponding values for the partial pressures of water-vapor above aqua ammonia in pounds per sq. in. abs., calculated from Goodenough's data by means of Equation (4), Chapter III, Bull. 146. Subtraction of these values from those of Table 38 has furnished the values of Table 40 as the partial pressures of ammonia in the vapor phase above the designated concentrations of aqua ammonia.

The Absorption Cycle.—The complete cycle of operation of the absorption refrigerating machine is illustrated diagrammatically by Fig. 45. Before proceeding with the operation of the complete cycle, it is well to note just what things are determining factors in the cycle.

The condenser operates in the same manner as in the compression system. The temperature of the condenser water, in the most part, determines the condenser pressure, while the quantity of water and amount of condenser surface also have probably a lesser important bearing on the magnitude of the pressure. The temperature of the water rises a few degrees in passing through the condenser, and there is a small temperature difference between the water leaving the condenser and the temperature of the condensing ammonia in the saturated portion of the condenser.

The temperature of the evaporating fluid in the evaporator is determined by the temperature desired in the cold storage or in the cold brine. The depression of the temperature is also affected by the amount of heat transmitting surface which is used. The temperature of the evaporator must always be a few degrees below the refrigerator. The pressures, of course, correspond to the various boiling temperatures of the fluid in the evaporator.

In a like manner, the temperature of the water available for cooling the absorber affects the conditions in the absorber. It is evident that the absorber must be supplied with cooling water, since the changes of states such as condensation and absorption are accompanied by a liberation of heat. With consideration to the relative amount of heat transmitting surface in the absorber, it is evident that the temperature of the cooling water will determine the temperature of the aqua in it. The temperature of the aqua must always be a few degrees above the water temperature, so that the heat of condensation and absorption of the ammonia vapor will flow into the cooling water. Thus, since the pressure is approximately the same as that of the evaporator, the exact condition of the aqua may be determined at once. Knowing the temperature and pressure of the aqua ammonia, the concentration may be

found by consulting Table 18 of Chapter III, or by calculation by means of formula (1) of this chapter.

In similar manner, the pressure in the generator is determined by the condenser pressure, which in turn depends upon the condenser water temperature. Also, it is obvious that heat must be supplied to distill or dissociate the ammonia from the aqua in the generator. This is generally supplied by condensing steam, and the temperature of the steam must be a few degrees above the temperature of the aqua in order to cause the heat to flow into the aqua, thereby distilling off the ammonia vapor and some water vapor. Hence, since the pressure and temperature are easily determined, the concentration may be found readily.

From the above, it will be noted that the temperature of the condenser water and the temperature that is desired in the refrigerator are the principal determining factors.

Assuming a condenser water temperature of 70° F. and refrigerator temperature of 20° F. the operation of the complete cycle may be noted. Beginning with the condenser, it will be further assumed that the condenser contains no moisture; that is, the condenser contains only anhydrous ammonia. The following tabulation will show the condition of pressures and temperatures in the condenser:

1. Water inlet temperature	70° F.
2. Temperature range, assumed	10° F.
3. Water outlet temperature	80° F.
4. Difference between water outlet and ammonia temp.	6° F.
5. Temperature of saturated ammonia	86° F.
6. Pressure of ammonia, lbs. abs.	170
7. Temperature liquid ammonia leaving condenser	86° F.

By referring to Fig. 45, it will be noted that the liquid ammonia at 86° F. and 170 lbs. pressure flows to the expansion valve, as in the compression system. The expansion valve throttles the pressure to that of the evaporator.

The conditions in the evaporator may be tabulated as follows:

1. Temperature of refrigerator	20° F.
2. Temperature difference, assumed	20° F.
3. Temperature of boiling ammonia	0° F.
4. Pressure of boiling ammonia, abs., lbs.	30
5. Temperature of vapor leaving evaporator	0° F.

Thus, the expansion valve throttles the pressure from 170 to 30 lbs.; the throttled liquid passes into the evaporator, where it is evaporated by absorbing its latent heat from the refrigerator; it leaves the evaporator and passes to the absorber at 0° F. and 30 lbs. pressure.

As the vapor passes into the absorber, it mixes with the weak aqua ammonia and is readily absorbed or it is dissolved quickly, due to the great affinity of the water for the ammonia. The weak aqua ammonia

absorbs all the ammonia that it can hold under the condition, and is then ready to be taken from the absorber, thereby removing the vapor from the evaporator.

The following tabulation indicates the conditions existing in the absorber, assuming that the absorber is cooled by water from the condenser:

1. Water inlet temperature	80° F.
2. Temperature range, assumed	10° F.
3. Water outlet temperature	90° F.
4. Temperature difference	8° F.
5. Temperature of aqua	98° F.
6. Pressure in absorber, lbs. abs.	30
7. Concentration, per cent	36
8. Temperature aqua leaving absorber	98° F.

The strong aqua of 36 per cent concentration at 98° F. and 30 lbs. pressure flows to the aqua pump, which discharges the strong aqua through the exchanger into the generator. The aqua pump is operated by mechanical power and simply increases the pressure from the low pressure of the evaporator and absorber to the high pressure of the generator and condenser. It is noted that this pump is the only piece of moving machinery in the whole system.

The conditions in the generator are shown in the following tabulation, assuming that the weak aqua is 10 per cent lower than the strong aqua, and that the aqua in the generator is 20° F. below the temperature of the condensing steam in the generator heating coils:

1. Generator pressure, same as condenser, lbs. abs.	170
2. Concentration of strong aqua	36%
3. Difference of concentration	10%
4. Concentration of weak aqua	26%
5. Temperature of aqua	243° F.
6. Temperature difference	20° F.
7. Temperature of steam	263° F.
8. Pressure of steam, lbs. abs.	37
9. Temperature of vapors leaving generator	243° F
10. Temperature of weak aqua leaving generator	243° F.

Thus, the generator receives the 36 per cent strong aqua from the absorber and distills off the ammonia until the solution contains only 26 per cent ammonia. This hot weak aqua is returned to the absorber after passing through the exchanger and weak aqua cooler. A difference of 8 to 10 per cent in the strength of the strong aqua and weak will produce economical operation of the system under ordinary conditions.

From the foregoing, it will be observed that the hot weak aqua from the generator and the cool strong aqua from the absorber are led through the heat exchanger. The function of the exchanger is to allow the heats of the strong aqua and weak aqua to interchange, since the weak aqua must be cooled to the absorber temperature and the strong

aqua must be heated nearly to the generator temperature. These liquids should flow through the exchanger in opposite directions for best results. The strong aqua may be assumed to be heated to its boiling temperature under the conditions, but no evaporation is allowed to

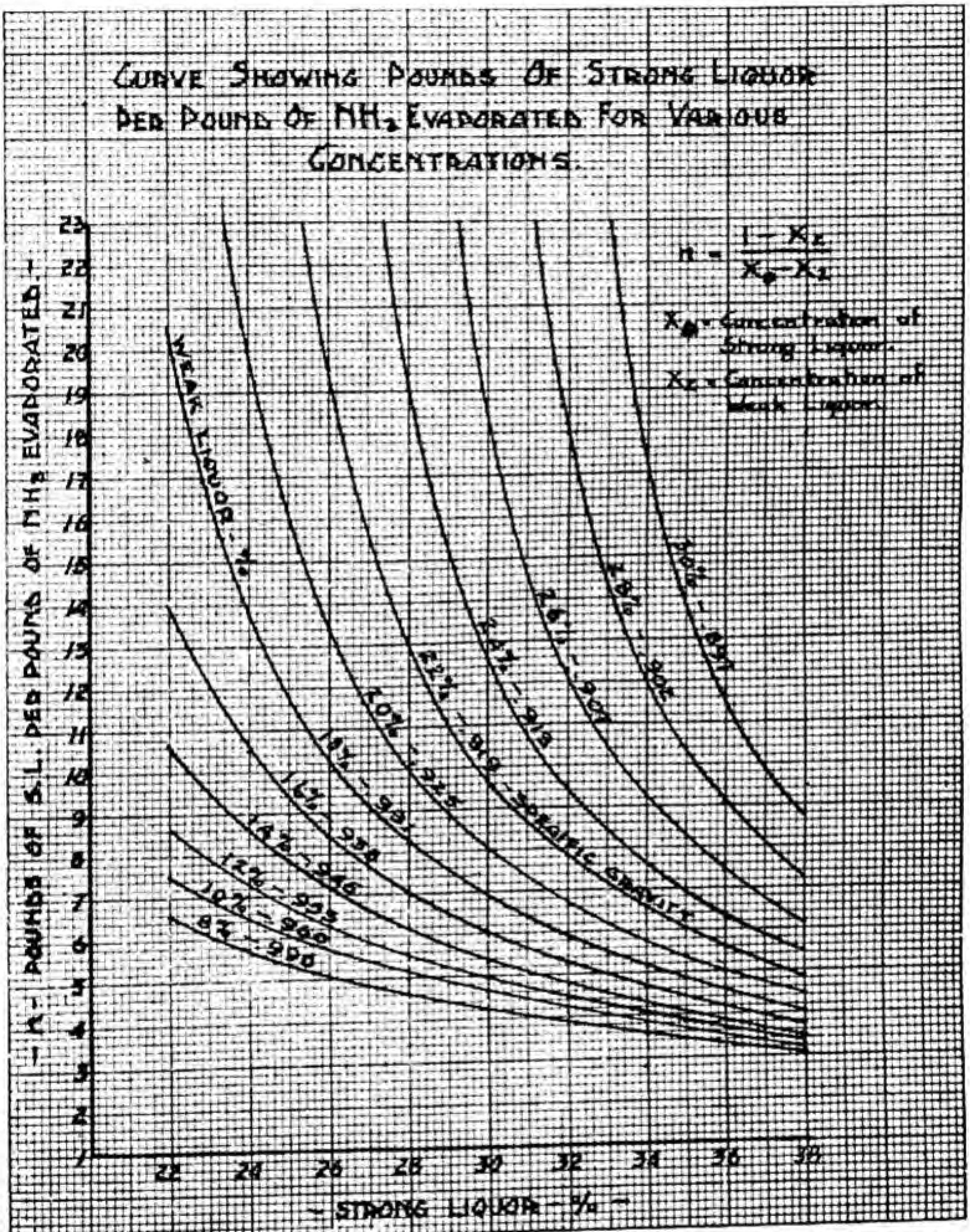


Fig. 46.—Chart Showing Pounds of Strong Liquor Per Pound of NH_3 Evaporated for Various Concentrations.

occur. The boiling temperature of 36 per cent strong aqua at 170 lbs. from Table 18 of Chapter III is equal to 203° F. approximately, so that the strong aqua may be heated to 203° F. in passing through the exchanger.

In order to calculate the drop of temperature of the weak aqua in passing through the exchanger, it is necessary to know the relative weights of the strong and weak aqua per pound of ammonia carried from the evaporator to the condenser.

The following formula indicates the amount of strong aqua to be pumped per pound of ammonia transferred:

$$N_1 = \frac{1 - X_2}{X_1 - X_2} \tag{4}$$

where N_1 = pounds of strong aqua
X_1 = concentration of strong aqua, expressed as a decimal
X_2 = concentration of weak aqua, expressed as a decimal

Fig. 46 has been plotted by H. G. Venemann to graphically give the number of pounds of strong aqua to be pumped per pound of ammonia transferred.

The pounds of weak aqua, N_2, pumped per lb. of ammonia transferred from the evaporator to the condenser, is equal to:

$$N_2 = N_1 - 1.$$

For the case of 36% and 26% aqua, since 36% = 0.36 and 26% = 0.26, the pounds of strong aqua to be circulated may be calculated as follows:

$$N_1 = \frac{1 - 0.26}{0.36 - 0.26} = \frac{0.74}{0.10} = 7.4 \text{ lbs}$$

$$N_2 = 7.4 - 1 = 6.4 \text{ lbs.}$$

Thus, it is evident that 7.4 lbs. of strong aqua pass through the exchanger in one direction, while 6.4 lbs. of weak aqua pass in the opposite direction. Assuming that there are no heat losses, or that the heat absorbed by the cool strong aqua is equal to the heat given up by the hot weak aqua, the temperature (t) of the weak aqua leaving the exchanger may be calculated as follows:

$$7.4\ (202^\circ - 98^\circ) = 6.4\ (243^\circ - t)$$

where t = temperature of weak aqua out of exchanger
thus t = 123°

The heat absorbed by the strong aqua is equal to 7.4 (202° − 98°), while that absorbed by the weak aqua is equal to 6.4 (243° − 123°). The specific heat of the aqua may be taken as 1.0 Btu. per lb.

The conditions existing in the exchanger may be tabulated as follows:

1. Temperature of strong aqua at inlet.......................... 98° F.
2. Temperature of strong aqua at outlet.......................... 202° F.
3. Temperature of weak aqua at inlet.......................... 243° F.
4. Temperature of weak aqua at outlet.......................... 123° F.
5. Pressure of strong and weak aqua lbs., abs.......................... 170

The weak aqua leaves the exchanger and passes through the weak aqua cooler and then through the regulating valve to the absorber. In passing through the weak aqua cooler, the aqua may be further cooled to 98° F., the temperature of the aqua in the absorber. It is also evident that the flow of the weak aqua from the generator into the absorber must be regulated in some manner. This is generally accomplished by means of a float-valve, which controls the height of the aqua in the absorber.

The ammonia and water vapor leaving the generator pass through the rectifier and then into the condenser. The function of the rectifier is to condense out the water vapor as much as possible, so that only anhydrous ammonia vapor goes to the condenser. The rectifier may be cooled by water or other means. The water vapor that is condensed out, of course, absorbs ammonia, making strong aqua, which must be trapped back to the generator. In order to be sure that none of the ammonia vapor condenses in the rectifier, the ammonia vapor leaving the rectifier should have a few degrees of superheat. The superheat may vary from 10° to 30° F. Assuming that 20° F. of superheat would be satisfactory, the temperature of the ammonia vapor leaving the rectifier would be $86° + 20° = 106°$ F. The temperature of the vapor entering the rectifier would be that of the vapors leaving the generator, approximately. Thus, the 106° F. and 170 lbs. pressure ammonia vapor passes to the condenser to be liquefied.

It is evident that the cycle is now completed and that the same anhydrous ammonia and the same aqua ammonia may be used over and over again.

Practical Absorption Systems.—Fig. 45 illustrates in a diagrammatic manner the principles of operation of the various important parts of the absorption refrigerating machine. The parts of apparatus, of course, never have such forms as are illustrated in Fig. 45. This figure was developed especially to show the principle underlying the operation of the various parts of the apparatus. Also, on further consideration, it is apparent that other parts of minor importance and further details must be added to those represented by Fig. 45, in order to comprise a commercially practical refrigerating system.

Hence the actual absorption machine may have a form such as indicated by Fig. 47. This is a diagram of the "Vogt" absorption refrigerating machine. The construction represented by Fig. 47 is known as the tubular construction, in contradistinction to the double-pipe and atmospheric types of systems.

Beginning with the condenser, it will be observed that the anhydrous liquid ammonia drains into a storage tank, which is generally termed the receiver. From this liquid receiver the ammonia is led to

the evaporator, which is, in this case, a brine cooler. After passing through the expansion valve or throttle valve, the ammonia enters the brine cooler and then after absorption of its latent heat of evaporation by cooling the brine, it passes to the bottom of the absorber in the form of a vapor or gas.

In the absorber, the vapor is absorbed by the solution, or aqua ammonia, giving up its latent heat of condensation and absorption. This heat is taken up by the cooling water.

The strong aqua, resulting from the absorption of the vapor from the brine cooler, is now pumped from the absorber into the bottom of the rectifier. After passing through the rectifier, it is led to the top of the exchanger, and after passing downward through the exchanger, it is finally led into the generator.

In the generator, the steam heats the ammonia solution, thereby distilling ammonia gas and steam from the aqua. The resultant weak aqua settles to the bottom of the generator, from which it is led to the bottom of the exchanger. Passing upward through the exchanger, it leaves at the top and is then piped to the bottom of the weak aqua cooler. Cooling water enters the weak aqua cooler at the top and leaves at the bottom thereby cooling the weak aqua to the desired temperature before it enters the absorber. From the weak aqua cooler, the weak aqua is now led through the regulating valve into the absorber to reabsorb ammonia vapor from the evaporator again.

In the meantime, the ammonia gases and steam, which have been distilled from the generator, pass to the rectifier. The strong aqua in counter-current to the gases, absorbs some heat from the gases, thereby condensing out the steam. The rectifier thus has a function similar to that of a regular steam condenser. The moisture that is removed from the gases is separated from the gases by means of a separator, and the resulting moisture is returned to the generator as strong aqua. The warm ammonia gas, now in the anhydrous state, passes to the condenser. Here, the cooling water absorbs the sensible heat and the latent heat of condensation, thereby liquefying the ammonia. The anhydrous liquid ammonia is now ready to be returned to the evaporator to reabsorb heat again. The anhydrous ammonia phase and the aqua ammonia phase of the cycle are thus completed, and the same cycle of operation may be repeated indefinitely.

The foregoing considerations illustrate the principal features of a commercially practical system.

Additional Details of Complete System.—According to the plant conditions and manufacturers of the apparatus, the general arrangements and methods of connecting the various parts of the apparatus will vary considerably. For instance, some manufacturers recommend

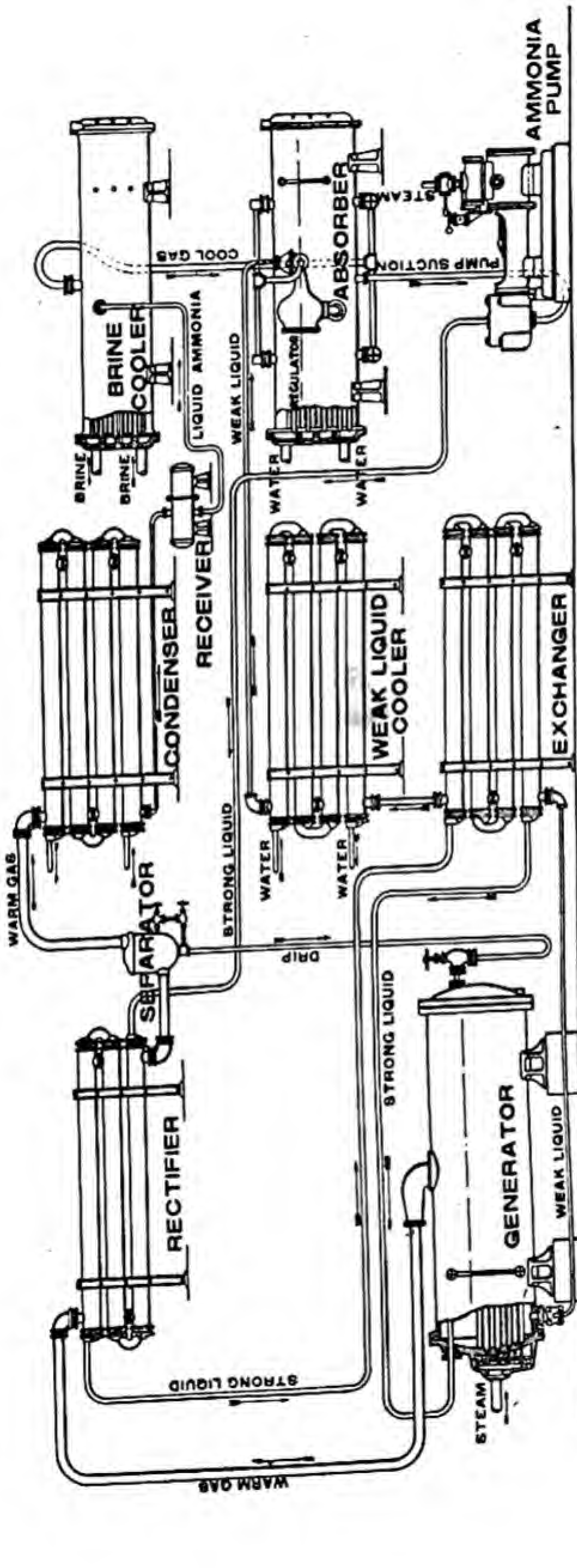


Fig. 47.—Vogt Tubular Absorption Machine.

the use of the analyzer, while some do not. The function of the analyzer is to further the heat interchange between the gases and the strong aqua ammonia. Analyzers were used frequently in the past, but, at present, they are to be used only under special conditions, generally.

Likewise, the manner of the circulation of the cooling water through the system demands special attention. In general, each installation is an individual problem, that must have its particular solution. However, under ordinary conditions, in plants having low pressures in the absorber and evaporator, it is probably best to pass all the cooling water through the absorber first and then through the condenser, after which the water may be used in the rectifier or weak aqua cooler. In high pressure plants, it may be desirable to pass all of the water through the condenser and then all or part through the absorber, after which the water may then be used in the rectifier and in the weak aqua cooler.

Relative to minor details of the system, it is evident that thermometers and pressure gauges must be installed at points where it is desirable to determine the conditions of pressures and temperatures. These details, which are of minor importance, are very helpful in maintaining the system in proper operation.

Likewise, gauge glasses should be provided at all points of the system where it is desirable to ascertain the level of the working liquids.

Purge valves and connections should be located at all points where it is desirable to remove air, gases or liquids from the parts of the apparatus for any purpose.

Suitable connections for charging aqua ammonia or anhydrous liquid into the system should be provided. Likewise, suitable connections for withdrawing strong and weak aqua for testing purposes should be provided.

General Arrangement of Complete Systems.—The general arrangement of the "Carbondale" double-pipe absorption refrigerating machine is shown by Fig. 48. In this system it will be observed that the anhydrous liquid ammonia flows to the receiver and then through an expansion valve into a shell and coil brine cooler. The vapor from the brine cooler flows to the absorber. The resultant strong aqua is led from the absorber through an aqua storage tank to the aqua ammonia pump. Leaving the aqua pump the strong aqua is discharged through the exchanger into the generator. The weak aqua ammonia, in the meantime, flows through the exchanger and weak aqua cooler to the absorber. The ammonia gas from the generator is led through the rectifier into the condenser. The cooling water flows through the

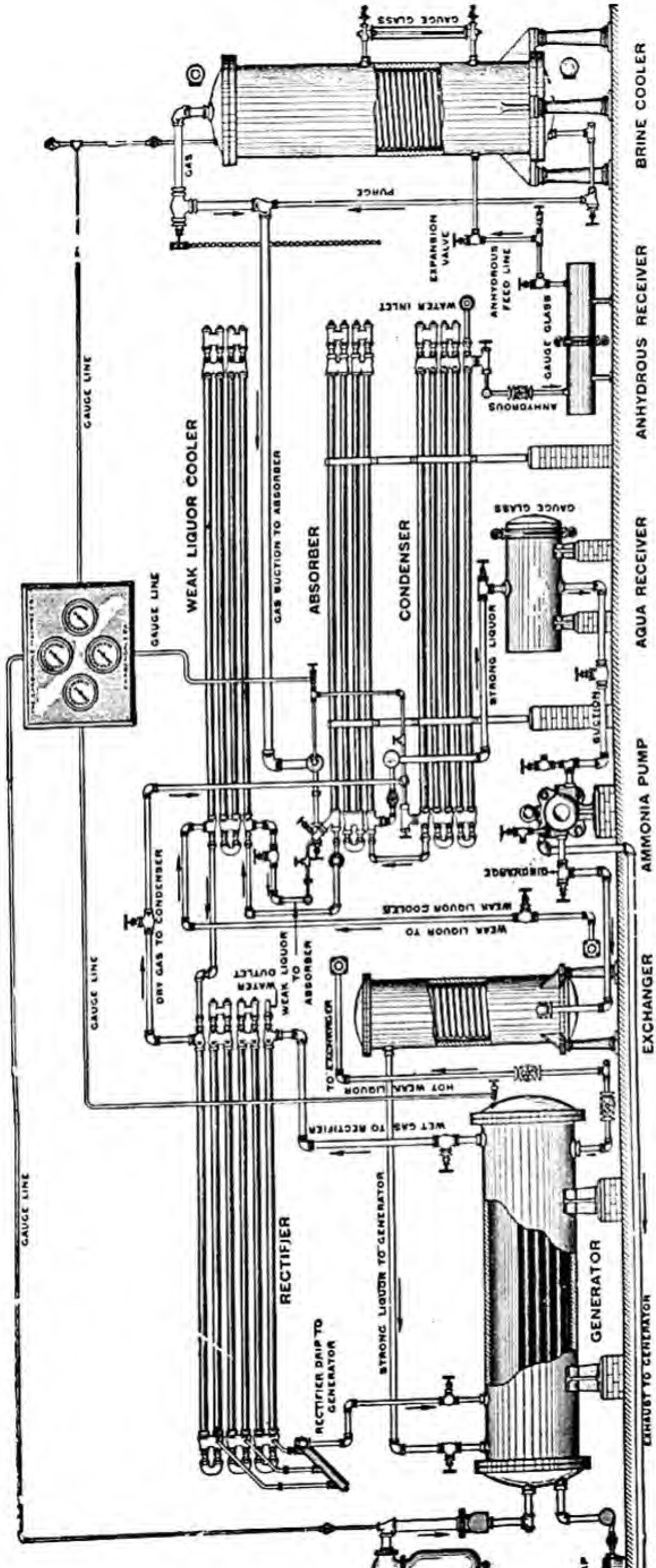


Fig. 48.—Carbondale Double-Pipe Absorption Machine.

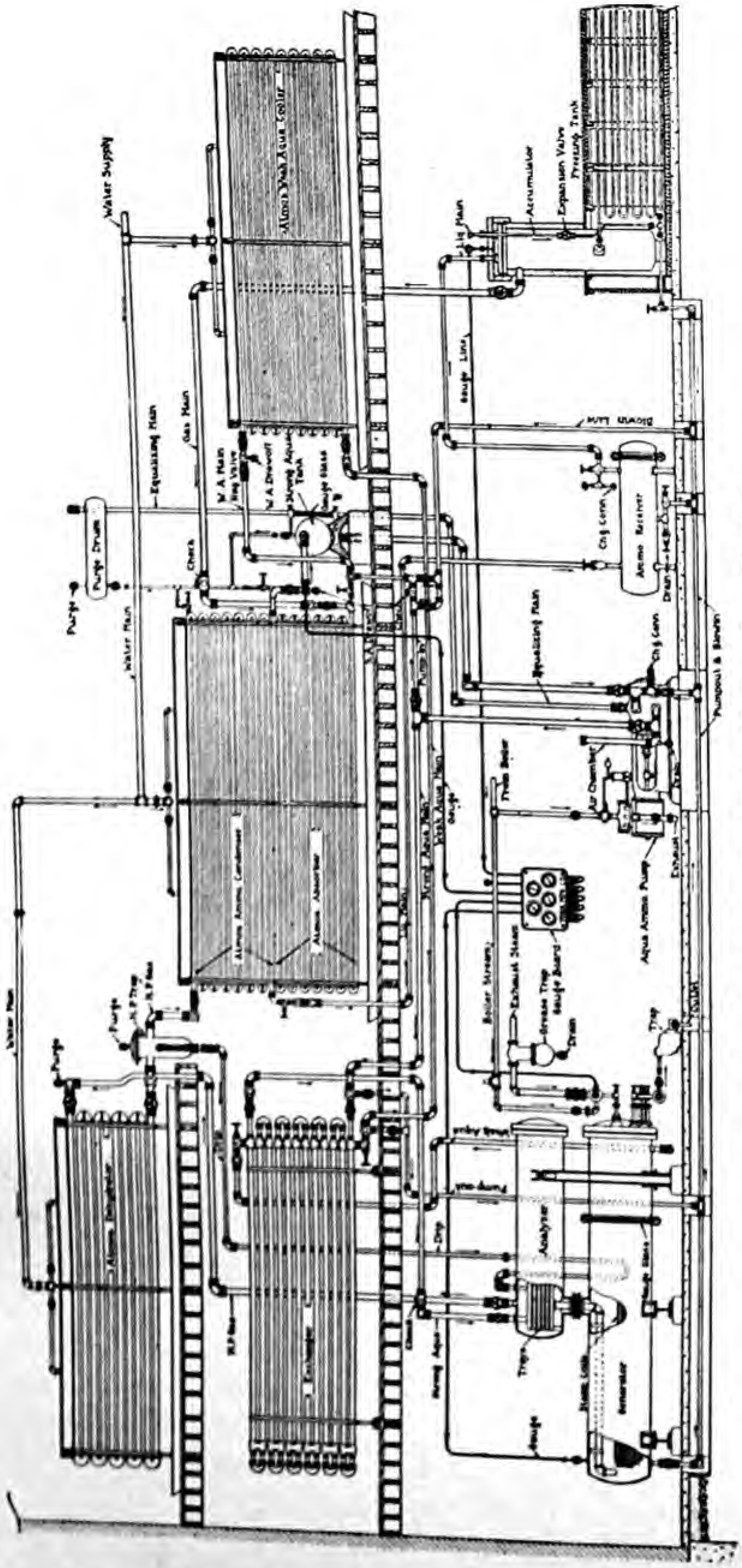


Fig. 49.—York Atmospheric Absorption Machine.

condenser, absorber, weak aqua cooler, and rectifier, to the sewer. Various pressure gauges, gauge glasses, connections, etc., are provided.

In a similar manner, the general arrangement of the "York" atmospheric type absorption machine is illustrated by Fig. 49. In this system, the anhydrous liquid ammonia leaving the condenser, passes through the receiver into the liquid cooling coil in the accumulator. The accumulator is connected to the direct expansion coils in the ice freezing tank. The liquid ammonia, after passing through the cooling coil, passes through the expansion valve into the evaporating coils. The vapor of the ammonia is led from the accumulator through a check valve into the bottom of the absorber. The resultant strong aqua leaves the absorber at the top and passes through the aqua storage tank to the aqua pump. The aqua pump discharges the strong aqua through the exchanger and analyzer into the generator. The weak aqua is led through the exchanger and weak aqua cooler to the absorber. The ammonia gas from the generator is led through the analyzer, the rectifier, or dehydrator, and the trap, or separator, into the condenser. The cooling water is divided, a small part going to the weak aqua cooler, a large part going over the condenser and then over the absorber, and another small part going over the dehydrator. Various pressure gauges and gauge glasses are provided. Purging, pump-out, pump-in, blow-in, gauge, equalizing, and steam connections are provided.

Practical Considerations.—From the foregoing considerations, it is apparent that the design as well as the operation of the various parts of the absorption refrigerating machine depend upon many factors, and while the whole system seems more or less complicated, it is comparatively simple when the principles underlying the operation of each piece of apparatus are thoroughly understood.

In the design, as well as the successful operation of the absorption refrigerating machine, it is further evident that detailed consideration must be given to the action of each piece of apparatus with respect to itself as well as the system as a whole. There is no difficulty in obtaining proper results with this type of system if the various pieces of apparatus are properly proportioned, and if the conditions in the plant correspond to those for which the apparatus was designed.

General Operating Conditions for Absorption Machines.—The plant conditions which determine the operating characteristics of absorption refrigerating machines when installed in cold storage plants are those of the temperature in the cold storage rooms, temperature of the brine, temperature of the cooling water, etc.

To assist in the determination of the operating conditions Figs. 50, 51, 52, 53 and 54 have been incorporated into the text. Fig. 50 gives

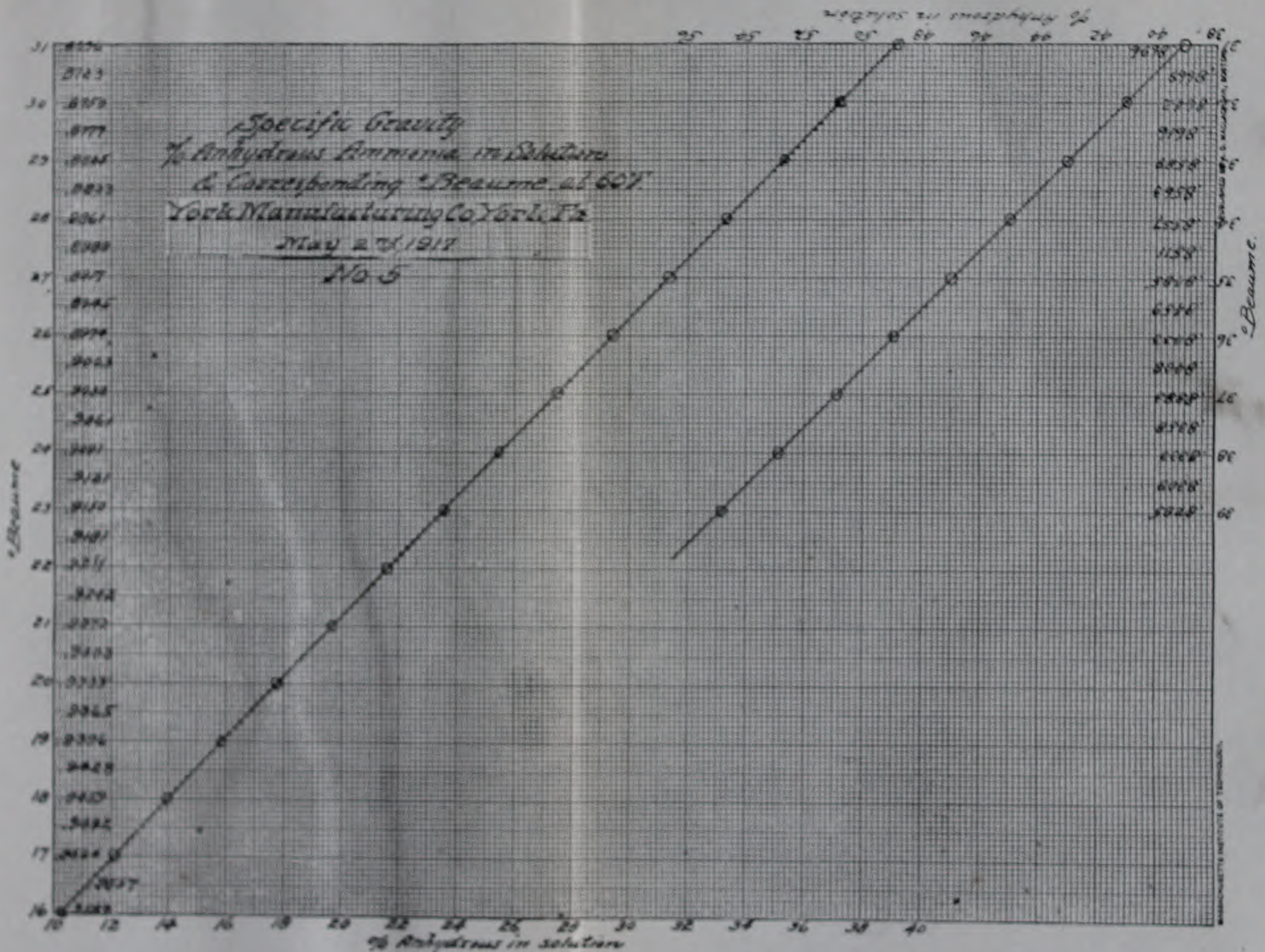


Fig. 50.—Specific Gravity of Aqua Ammonia.

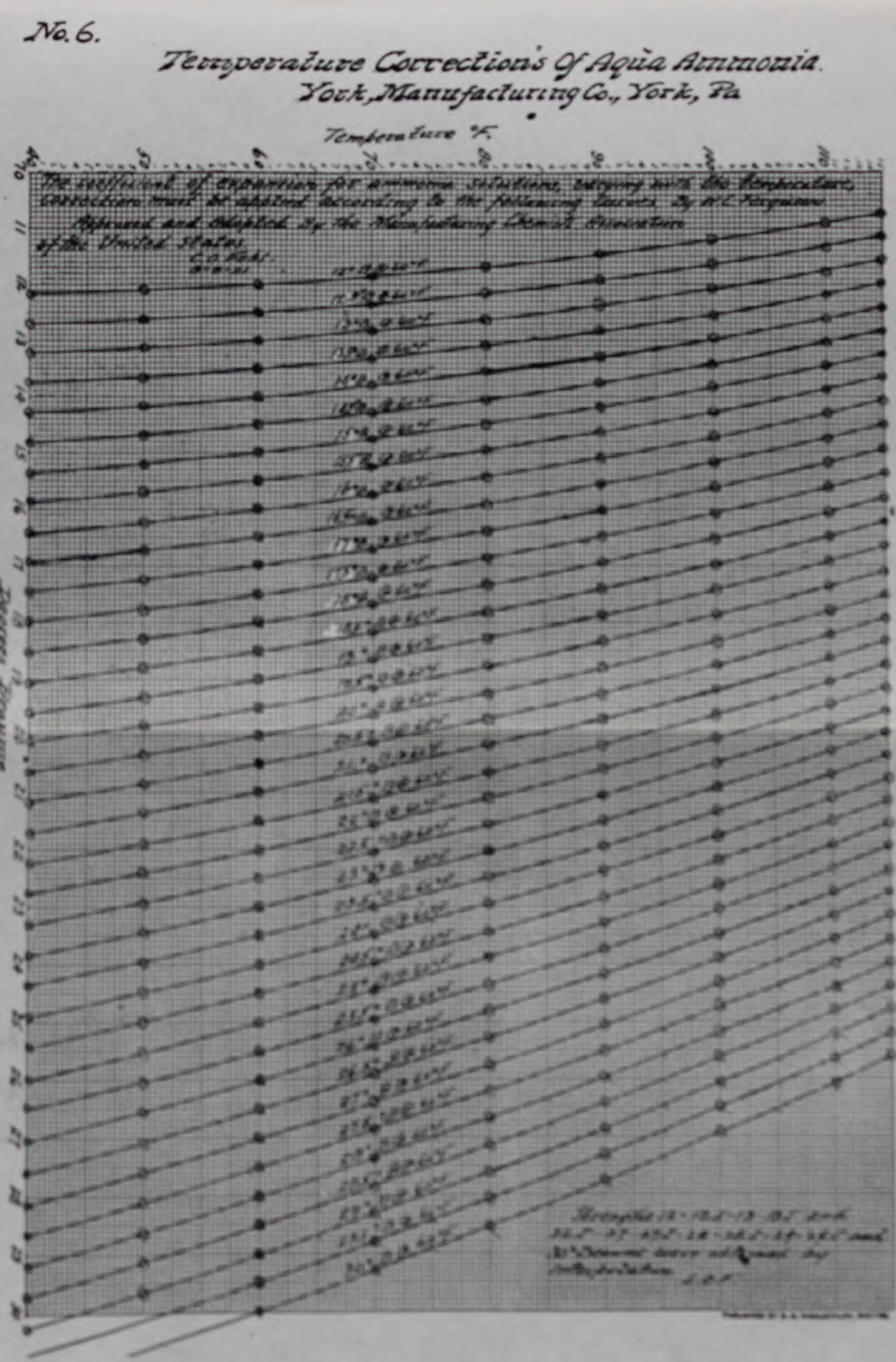


Fig. 51.—Temperature Corrections for Aqua Ammonia.

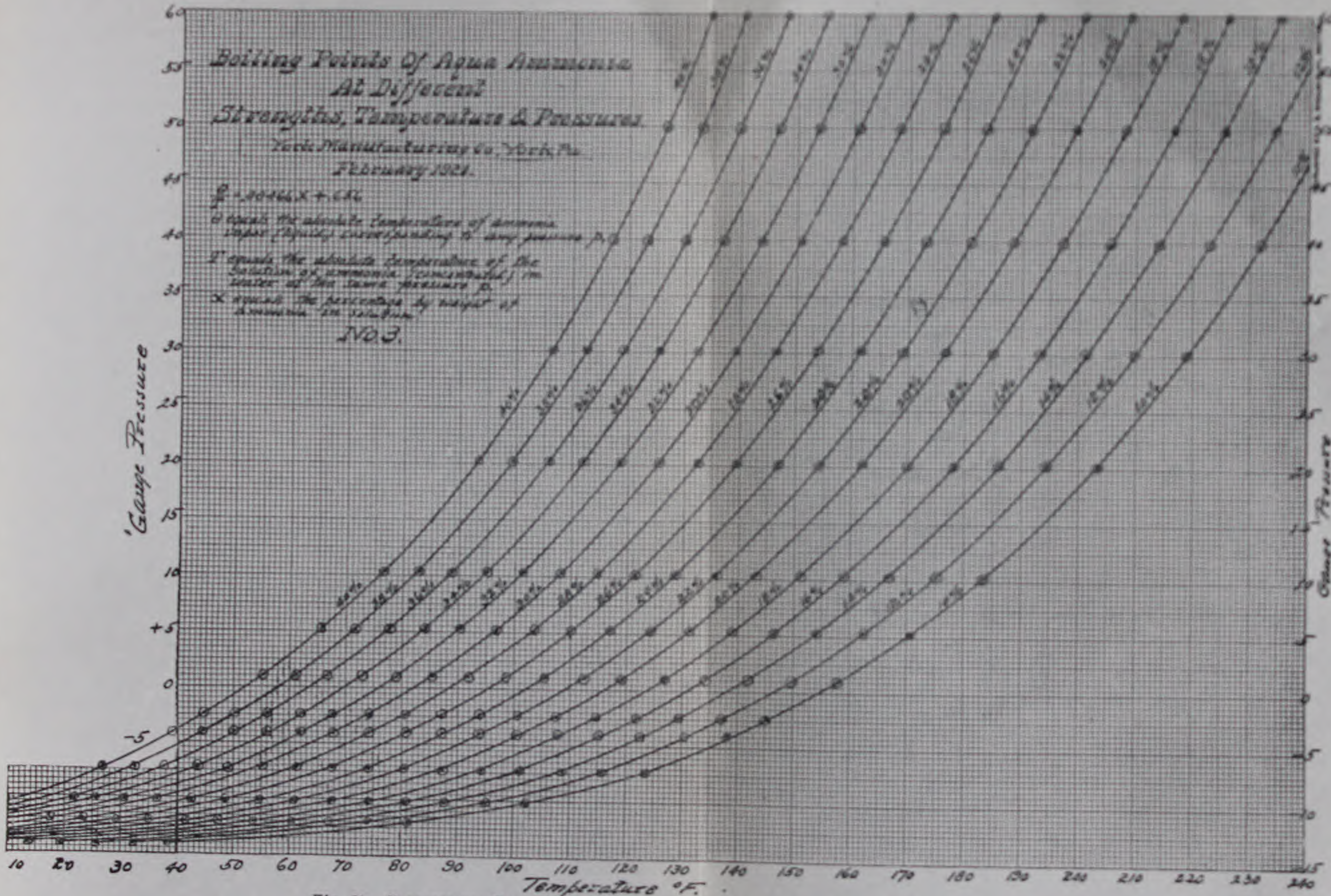


Fig. 52.—Temperature Pressure Concentration Relation of Low Strength Aqua Ammonia.

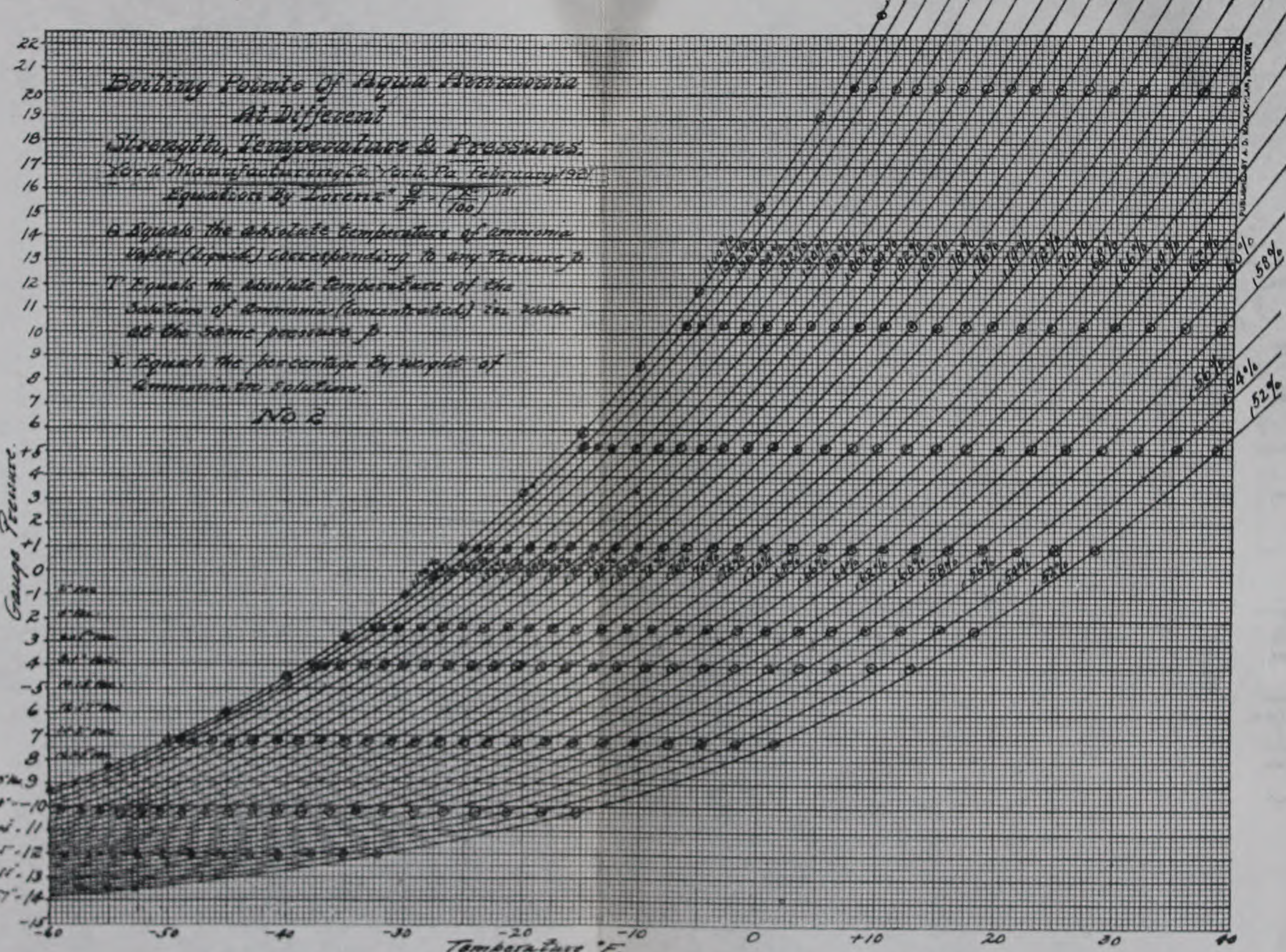


Fig. 53.—Temperature Pressure Concentration Relation of High Strength Aqua Ammonia.

the specific gravity and degrees Beaumé which correspond to various percentages of ammonia in a solution of aqua ammonia. These specific gravities have been taken at 60° F., and compared to distilled water at 60° F. Since specific gravities may be determined at temperatures other than 60° F., and since the specific gravities thus determined would be affected by the change of volume due to the change of temperature, corrections must be made for the expansion of the liquid. These corrections are indicated by Fig. 51. These give the corrections that must be applied to aqua ammonia having varying strengths and varying temperatures. From Fig. 51 it will be noted that when the temperature of the aqua ammonia is below 60° F. a correction must be added to the reading; likewise, when the temperature is above 60° F. a correction must be substracted from the reading.

Fig. 52 shows the temperature, pressure and concentration relations of aqua ammonia when the concentration varies from 10 to 40 per cent. This figure shows graphically most of the values given in Table 18 of Chapter III. It should be noted that the pressures shown in Fig. 52 are gauge pressures. Fig. 53 shows the pressure, temperature, concentration relation of aqua ammonia having strengths varying from 52 to 100 per cent. Fig. 53 will also be found useful in determining the various operating conditions in the absorption refrigerating machine. Fig. 54 shows the temperature at which the ammonia gas should leave the dehydrator in order to make sure that no condensation takes place in the dehydrator.

Figs. 50, 51, 52, 53 and 54 have been furnished to the author through the courtesy of the late Thomas Shipley, formerly vice-president and general manager of the York Corporation, York, Pa. The work of preparing these diagrams was done under the supervision of Mr. C. O. Fehl, absorption engineer with the York Corporation.

The absorption refrigerating machine has its own particular field of application. It operates quite economically at low evaporator pressures. At evaporator pressures below eight to ten pounds gauge suction pressure, the absorption machine will have an economy greater than the ammonia compression refrigerating machine driven by compound condensing steam engine.

The continued increase in the application of the two-stage ammonia compression system for lower evaporator pressures is infringing upon the field of the application of the absorption machine. However, in plants where there is a quantity of low pressure steam available, it may be advantageous to install the absorption machine. As previously indicated, when exceptionally low temperatures are desired, the absorption refrigerating machine will produce these lower temperatures in a more economical manner.

TABLE 41.—PROPERTIES OF SATURATED STEAM.
Marks and Davis (LONGMANS, GREEN & CO.)

Vacuum, Inches of Mercury.	Absolute Pressure, Lbs. per Sq. In.	Temperature, Fahrenheit.	Total Heat above 32° F.		Latent Heat, $L = H - h$ Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam.	Weight of 1 Cu. Ft. Steam, Lb.	Entropy of the Water.	Entropy of Evaporation.
			In the Water h Heat-Units.	In the Steam H Heat-Units.					
29.74	0.0886	32	0.00	1073.4	1073.4	3294	0.000304	0.0000	2.1832
29.67	0.1217	40	8.05	1076.9	1068.9	2438	0.000410	0.0162	2.1394
29.56	0.1780	50	18.08	1081.4	1063.3	1702	0.000587	0.0361	2.0865
29.40	0.2562	60	28.08	1085.9	1057.8	1208	0.000828	0.0555	2.0358
29.18	0.3626	70	38.06	1090.3	1052.3	871	0.001148	0.0745	1.9868
28.89	0.505	80	48.03	1094.8	1046.7	636.8	0.001570	0.0932	1.9398
28.50	0.696	90	58.00	1099.2	1041.2	469.3	0.002131	0.1114	1.8944
28.00	0.946	100	67.97	1103.6	1035.6	350.8	0.002851	0.1295	1.8505
27.88	1	101.83	69.8	1104.4	1034.6	333.0	0.00300	0.1327	1.8427
25.85	2	126.15	94.0	1115.0	1021.0	173.5	0.00576	0.1749	1.7431
23.81	3	141.52	109.4	1121.6	1012.3	118.5	0.00845	0.2008	1.6840
21.78	4	153.01	120.9	1126.5	1005.7	90.5	0.01107	0.2198	1.6416
19.74	5	162.28	130.1	1130.5	1000.3	73.33	0.01364	0.2348	1.6084
17.70	6	170.06	137.9	1133.7	995.8	61.89	0.01616	0.2471	1.5814
15.67	7	176.85	144.7	1136.5	991.8	53.56	0.01867	0.2579	1.5582
13.63	8	182.86	150.8	1139.0	988.2	47.27	0.02115	0.2673	1.5380
11.60	9	188.27	156.2	1141.1	985.0	42.36	0.02361	0.2756	1.5202
9.56	10	193.22	161.1	1143.1	982.0	38.38	0.02606	0.2832	1.5042
7.52	11	197.75	165.7	1144.9	979.2	35.10	0.02849	0.2902	1.4895
5.49	12	201.96	169.9	1146.5	976.6	32.36	0.03090	0.2967	1.4760
3.45	13	205.87	173.8	1148.0	974.2	30.03	0.03330	0.3025	1.4639
1.42	14	209.55	177.5	1149.4	971.9	28.02	0.03569	0.3081	1.4523
lbs. gage.	14.70	212	180.0	1150.4	970.4	26.79	0.03732	0.3118	1.4447
0.3	15	213.0	181.0	1150.7	969.7	26.27	0.03806	0.3133	1.4416
1.3	16	216.3	184.4	1152.0	967.6	24.79	0.04042	0.3183	1.4311
2.3	17	219.4	187.5	1153.1	965.6	23.38	0.04277	0.3229	1.4215
3.3	18	222.4	190.5	1154.2	963.7	22.16	0.04512	0.3273	1.4127
4.3	19	225.2	193.4	1155.2	961.8	21.07	0.04746	0.3315	1.4045
5.3	20	228.0	196.1	1156.2	960.0	20.08	0.04980	0.3355	1.3965
6.3	21	230.6	198.8	1157.1	958.3	19.18	0.05213	0.3393	1.3887
7.3	22	233.1	201.3	1158.0	956.7	18.37	0.05445	0.3430	1.3811
8.3	23	235.5	203.8	1158.8	955.1	17.62	0.05676	0.3465	1.3739
9.3	24	237.8	206.1	1159.6	953.5	16.93	0.05907	0.3499	1.3670
10.3	25	240.1	208.4	1160.4	952.0	16.30	0.0614	0.3532	1.3604
11.3	26	242.2	210.6	1161.2	950.6	15.72	0.0636	0.3564	1.3542
12.3	27	244.4	212.7	1161.9	949.2	15.18	0.0659	0.3594	1.3483
13.3	28	246.4	214.8	1162.6	947.8	14.67	0.0682	0.3623	1.3425
14.3	29	248.4	216.8	1163.2	946.4	14.19	0.0705	0.3652	1.3367
15.3	30	250.3	218.8	1163.9	945.1	13.74	0.0728	0.3680	1.3311
16.3	31	252.2	220.7	1164.5	943.8	13.32	0.0751	0.3707	1.3257
17.3	32	254.1	222.6	1165.1	942.5	12.93	0.0773	0.3733	1.3205
18.3	33	255.8	224.4	1165.7	941.3	12.57	0.0795	0.3759	1.3155
19.3	34	257.6	226.2	1166.3	940.1	12.22	0.0818	0.3784	1.3107
20.3	35	259.3	227.9	1166.8	938.9	11.89	0.0841	0.3808	1.3060
21.3	36	261.0	229.6	1167.3	937.7	11.58	0.0863	0.3832	1.3014
22.3	37	262.6	231.3	1167.8	936.6	11.29	0.0886	0.3855	1.2969
23.3	38	264.2	232.9	1168.4	935.5	11.01	0.0908	0.3877	1.2925
24.3	39	265.8	234.5	1168.9	934.4	10.74	0.0931	0.3899	1.2882
25.3	40	267.3	236.1	1169.4	933.3	10.49	0.0953	0.3920	1.2841
26.3	41	268.7	237.6	1169.8	932.2	10.25	0.0976	0.3941	1.2800

TABLE 41.—PROPERTIES OF SATURATED STEAM.—(Continued.)

Gauge Pressure, Lbs. per Sq. In.	Absolute Pressure, Lbs. per Sq. In.	Temperature, Fahrenheit.	Total Heat above 32° F. In the Water h Heat-Units.	Total Heat above 32° F. In the Steam H Heat-Units.	Latent Heat, $L = H - h$ Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam.	Weight of 1 Cu. Ft. Steam, Lb.	Entropy of the Water.	Entropy of Evaporation.
27.3	42	270.2	239.1	1170.3	931.2	10.02	0.0998	0.3962	1.2759
28.3	43	271.7	240.5	1170.7	930.2	9.80	0.1020	0.3982	1.2720
29.3	44	273.1	242.0	1171.2	929.2	9.59	0.1043	0.4002	1.2681
30.3	45	274.5	243.4	1171.6	928.2	9.39	0.1065	0.4021	1.2644
31.3	46	275.8	244.8	1172.0	927.2	9.20	0.1087	0.4040	1.2607
32.3	47	277.2	246.1	1172.4	926.3	9.02	0.1109	0.4059	1.2571
33.3	48	278.5	247.5	1172.8	925.3	8.84	0.1131	0.4077	1.2536
34.3	49	279.8	248.8	1173.2	924.4	8.67	0.1153	0.4095	1.2502
35.3	50	281.0	250.1	1173.6	923.5	8.51	0.1175	0.4113	1.2468
36.3	51	282.3	251.4	1174.0	922.6	8.35	0.1197	0.4130	1.2432
37.3	52	283.5	252.6	1174.3	921.7	8.20	0.1219	0.4147	1.2405
38.3	53	284.7	253.9	1174.7	920.8	8.05	0.1241	0.4164	1.2370
39.3	54	285.9	255.1	1175.0	919.9	7.91	0.1263	0.4180	1.2339
40.3	55	287.1	256.3	1175.4	919.0	7.78	0.1285	0.4196	1.2309
41.3	56	288.2	257.5	1175.7	918.2	7.65	0.1307	0.4212	1.2278
42.3	57	289.4	258.7	1176.0	917.4	7.52	0.1329	0.4227	1.2248
43.3	58	290.5	259.8	1176.4	916.5	7.40	0.1350	0.4242	1.2218
44.3	59	291.6	261.0	1176.7	915.7	7.28	0.1372	0.4257	1.2189
45.3	60	292.7	262.1	1177.0	914.9	7.17	0.1394	0.4272	1.2160
46.3	61	293.8	263.2	1177.3	914.1	7.06	0.1416	0.4287	1.2132
47.3	62	294.9	264.3	1177.6	913.3	6.95	0.1438	0.4302	1.2104
48.3	63	295.9	265.4	1177.9	912.5	6.85	0.1460	0.4316	1.2077
49.3	64	297.0	266.4	1178.2	911.8	6.75	0.1482	0.4330	1.2050
50.3	65	298.0	267.5	1178.5	911.0	6.65	0.1503	0.4344	1.2024
51.3	66	299.0	268.5	1178.8	910.2	6.56	0.1525	0.4358	1.1998
52.3	67	300.0	269.6	1179.0	909.5	6.47	0.1547	0.4371	1.1972
53.3	68	301.0	270.6	1179.3	908.7	6.38	0.1569	0.4385	1.1946
54.3	69	302.0	271.6	1179.6	908.0	6.29	0.1590	0.4398	1.1921
55.3	70	302.9	272.6	1179.8	907.2	6.20	0.1612	0.4411	1.1896
56.3	71	303.9	273.6	1180.1	906.5	6.12	0.1634	0.4424	1.1872
57.3	72	304.8	274.5	1180.4	905.8	6.04	0.1656	0.4437	1.1848
58.3	73	305.8	275.5	1180.6	905.1	5.96	0.1678	0.4449	1.1825
59.3	74	306.7	276.5	1180.9	904.4	5.89	0.1699	0.4462	1.1801
60.3	75	307.6	277.4	1181.1	903.7	5.81	0.1721	0.4474	1.1778
61.3	76	308.5	278.3	1181.4	903.0	5.74	0.1743	0.4487	1.1755
62.3	77	309.4	279.3	1181.6	902.3	5.67	0.1764	0.4499	1.1730
63.3	78	310.3	280.2	1181.8	901.7	5.60	0.1786	0.4511	1.1712
64.3	79	311.2	281.1	1182.1	901.0	5.54	0.1808	0.4523	1.1687
65.3	80	312.0	282.0	1182.3	900.3	5.47	0.1829	0.4535	1.1665
66.3	81	312.9	282.9	1182.5	899.7	5.41	0.1851	0.4546	1.1644
67.3	82	313.8	283.8	1182.8	899.0	5.34	0.1873	0.4557	1.1623
68.3	83	314.6	284.6	1183.0	898.4	5.28	0.1894	0.4568	1.1602
69.3	84	315.4	285.5	1183.2	897.7	5.22	0.1915	0.4579	1.1581
70.3	85	316.3	286.3	1183.4	897.1	5.16	0.1937	0.4590	1.1561
71.3	86	317.1	287.2	1183.6	896.4	5.10	0.1959	0.4601	1.1540
72.3	87	317.9	288.0	1183.8	895.8	5.05	0.1980	0.4612	1.1520
73.3	88	318.7	288.9	1184.0	895.2	5.00	0.2001	0.4623	1.1500
74.3	89	319.5	289.7	1184.2	894.6	4.94	0.2023	0.4633	1.1481
75.3	90	320.3	290.5	1184.4	893.9	4.89	0.2044	0.4644	1.1461
76.3	91	321.1	291.3	1184.6	893.3	4.84	0.2065	0.4654	1.1442
77.3	92	321.8	292.1	1184.8	892.7	4.79	0.2087	0.4664	1.1423
78.3	93	322.6	292.9	1185.0	892.1	4.74	0.2109	0.4674	1.1404
79.3	94	323.4	293.7	1185.2	891.5	4.69	0.2130	0.4684	1.1385
80.3	95	324.1	294.5	1185.4	890.9	4.65	0.2151	0.4694	1.1367

TABLE 41.—PROPERTIES OF SATURATED STEAM.—(Continued.)

Gauge Pressure, Lbs. per Sq. In.	Absolute Pressure, Lbs. per Sq. In.	Temperature, Fahrenheit.	Total Heat above 32° F.		Latent Heat, $L = H - h$ Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam.	Weight of 1 Cu. Ft. Steam, Lb.	Entropy of the Water.	Entropy of Evaporation.
			In the Water h Heat-Units.	In the Steam H Heat-Units.					
81.3	96	324.9	295.3	1185.6	890.3	4.60	0.2172	0.4704	1.1348
82.3	97	325.6	296.1	1185.8	889.7	4.56	0.2193	0.4714	1.1330
83.3	98	326.4	296.8	1186.0	889.2	4.51	0.2215	0.4724	1.1312
84.3	99	327.1	297.6	1186.2	888.6	4.47	0.2237	0.4733	1.1295
85.3	100	327.8	298.3	1186.3	888.0	4.429	0.2258	0.4743	1.1277
87.3	102	329.3	299.8	1186.7	886.9	4.347	0.2300	0.4762	1.1242
89.3	104	330.7	301.3	1187.0	885.8	4.268	0.2343	0.4780	1.1208
91.3	106	332.0	302.7	1187.4	884.7	4.192	0.2336	0.4798	1.1174
93.3	108	333.4	304.1	1187.7	883.6	4.118	0.2429	0.4816	1.1141
95.3	110	334.8	305.5	1188.0	882.5	4.047	0.2472	0.4834	1.1108
97.3	112	336.1	306.9	1188.4	881.4	3.978	0.2514	0.4852	1.1076
99.3	114	337.4	308.3	1188.7	880.4	3.912	0.2556	0.4869	1.1045
101.3	116	338.7	309.6	1189.0	879.3	3.848	0.2599	0.4886	1.1014
103.3	118	340.0	311.0	1189.3	878.3	3.786	0.2641	0.4903	1.0984
105.3	120	341.3	312.3	1189.6	877.2	3.726	0.2683	0.4919	1.0954
107.3	122	342.5	313.6	1189.8	876.2	3.668	0.2726	0.4935	1.0924
109.3	124	343.8	314.9	1190.1	875.2	3.611	0.2769	0.4951	1.0895
111.3	126	345.0	316.2	1190.4	874.2	3.556	0.2812	0.4967	1.0865
113.3	128	346.2	317.4	1190.7	873.3	3.504	0.2854	0.4982	1.0837
115.3	130	347.4	318.6	1191.0	872.3	3.452	0.2897	0.4998	1.0809
117.3	132	348.5	319.9	1191.2	871.3	3.402	0.2939	0.5013	1.0782
119.3	134	349.7	321.1	1191.5	870.4	3.354	0.2981	0.5028	1.0755
121.3	136	350.8	322.3	1191.7	869.4	3.308	0.3023	0.5043	1.0728
123.3	138	352.0	323.4	1192.0	868.5	3.263	0.3065	0.5057	1.0702
125.3	140	353.1	324.6	1192.2	867.6	3.219	0.3107	0.5072	1.0675
127.3	142	354.2	325.8	1192.5	866.7	3.175	0.3150	0.5086	1.0649
129.3	144	355.3	326.9	1192.7	865.8	3.133	0.3192	0.5100	1.0624
131.3	146	356.3	328.0	1192.9	864.9	3.092	0.3234	0.5114	1.0599
133.3	148	357.4	329.1	1193.2	864.0	3.052	0.3276	0.5128	1.0574
135.3	150	358.5	330.2	1193.4	863.2	3.012	0.3320	0.5142	1.0550
137.3	152	359.5	331.4	1193.6	862.3	2.974	0.3362	0.5155	1.0525
139.3	154	360.5	332.4	1193.8	861.4	2.938	0.3404	0.5169	1.0501
141.3	156	361.6	333.5	1194.1	860.6	2.902	0.3446	0.5182	1.0477
143.3	158	362.6	334.6	1194.3	859.7	2.868	0.3488	0.5195	1.0454
145.3	160	363.6	335.6	1194.5	858.8	2.834	0.3529	0.5208	1.0431
147.3	162	364.6	336.7	1194.7	858.0	2.801	0.3570	0.5220	1.0409
149.3	164	365.6	337.7	1194.9	857.2	2.769	0.3612	0.5233	1.0387
151.3	166	366.5	338.7	1195.1	856.4	2.737	0.3654	0.5245	1.0365
153.3	168	367.5	339.7	1195.3	855.5	2.706	0.3696	0.5257	1.0343
155.3	170	368.5	340.7	1195.4	854.7	2.675	0.3738	0.5269	1.0321
157.3	172	369.4	341.7	1195.6	853.9	2.645	0.3780	0.5281	1.0300
159.3	174	370.4	342.7	1195.8	853.1	2.616	0.3822	0.5293	1.0278
161.3	176	371.3	343.7	1196.0	852.3	2.588	0.3864	0.5305	1.0257
163.3	178	372.2	344.7	1196.2	851.5	2.560	0.3906	0.5317	1.0235
165.3	180	373.1	345.6	1196.4	850.8	2.533	0.3948	0.5328	1.0215
167.3	182	374.0	346.6	1196.6	850.0	2.507	0.3989	0.5339	1.0195
169.3	184	374.9	347.6	1196.8	849.2	2.481	0.4031	0.5351	1.0174
171.3	186	375.8	348.5	1196.9	848.4	2.455	0.4073	0.5362	1.0154
173.3	183	376.7	349.4	1197.1	847.7	2.430	0.4115	0.5373	1.0134
175.3	190	377.6	350.4	1197.3	846.9	2.406	0.4157	0.5384	1.0114
177.3	192	378.5	351.3	1197.4	846.1	2.381	0.4199	0.5395	1.0095
179.3	194	379.3	352.2	1197.6	845.4	2.358	0.4241	0.5405	1.0076
181.3	196	380.2	353.1	1197.8	844.7	2.335	0.4283	0.5416	1.0056
183.3	198	381.0	354.0	1197.9	843.9	2.312	0.4325	0.5426	1.0038

TABLE 41.—PROPERTIES OF SATURATED STEAM.—(Concluded.)

Gauge Pressure, Lbs. per Sq. In.	Absolute Pressure, Lbs. per Sq. In.	Temperature, Fahrenheit.	Total Heat above 32° F.		Latent Heat, $L = H - h$ Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam.	Weight of 1 Cu. Ft. Steam, Lb.	Entropy of the Water.	Entropy of Evaporation.
			In the Water h Heat-Units.	In the Steam H Heat-Units.					
185.3	200	381.9	354.9	1198.1	843.2	2.290	0.437	0.5437	1.0019
190.3	205	384.0	357.1	1198.5	841.4	2.237	0.447	0.5463	0.9973
195.3	210	386.0	359.2	1198.8	839.6	2.187	0.457	0.5488	0.9928
200.3	215	388.0	361.4	1199.2	837.9	2.138	0.468	0.5513	0.9885
205.3	220	389.9	363.4	1199.6	836.2	2.091	0.478	0.5538	0.9841
210.3	225	391.9	365.5	1199.9	834.4	2.046	0.489	0.5562	0.9799
215.3	230	393.8	367.5	1200.2	832.8	2.004	0.499	0.5586	0.9758
220.3	235	395.6	369.4	1200.6	831.1	1.964	0.509	0.5610	0.9717
225.3	240	397.4	371.4	1200.9	829.5	1.924	0.520	0.5633	0.9676
230.3	245	399.3	373.3	1201.2	827.9	1.887	0.530	0.5655	0.9638
235.3	250	401.1	375.2	1201.5	826.3	1.850	0.541	0.5676	0.9600
245.3	260	404.5	378.9	1202.1	823.1	1.782	0.561	0.5719	0.9525
255.3	270	407.9	382.5	1202.6	820.1	1.718	0.582	0.5760	0.9454
265.3	280	411.2	386.0	1203.1	817.1	1.658	0.603	0.5800	0.9385
275.3	290	414.4	389.4	1203.6	814.2	1.602	0.624	0.5840	0.9316
285.3	300	417.5	392.7	1204.1	811.3	1.551	0.645	0.5878	0.9251
295.3	310	420.5	395.9	1204.5	808.5	1.502	0.666	0.5915	0.9187
305.3	320	423.4	399.1	1204.9	805.8	1.456	0.687	0.5951	0.9125
315.3	330	426.3	402.2	1205.3	803.1	1.413	0.708	0.5986	0.9065
325.3	340	429.1	405.3	1205.7	800.4	1.372	0.729	0.6020	0.9006
335.3	350	431.9	408.2	1206.1	797.8	1.334	0.750	0.6053	0.8949
345.3	360	434.6	411.2	1206.4	795.3	1.298	0.770	0.6085	0.8894
355.3	370	437.2	414.0	1206.8	792.8	1.264	0.791	0.6116	0.8840
365.3	380	439.8	416.8	1207.1	790.3	1.231	0.812	0.6147	0.8788
375.3	390	442.3	419.5	1207.4	787.9	1.200	0.833	0.6178	0.8737
385.3	400	444.8	422	1208	786	1.17	0.86	0.621	0.868
435.3	450	456.5	435	1209	774	1.04	0.96	0.635	0.844
485.3	500	467.3	448	1210	762	0.93	1.08	0.648	0.822
535.3	550	477.3	459	1210	751	0.83	1.20	0.659	0.801
585.3	600	486.6	469	1210	741	0.76	1.32	0.670	0.783

QUESTIONS ON CHAPTER V.

1. What is the general principle underlying the operation of the absorption part of the absorption refrigerating machine?

2. How do the properties of a solution of ammonia and water vary when the pressure on the solution is held constant?

3. Explain how the properties of a solution vary when the temperature of the solution is held constant.

4. Name the ten principal parts of the elementary absorption machine.

5. Describe briefly the function of each of the ten principal parts of the elementary absorption machine.

6. How much does a gallon of 26° Beaumé aqua ammonia weigh at 60° F., if a gallon of water weighs 8.33 pounds. What are the respective weights of water and ammonia per gallon under this condition?

7. If it is desired to maintain a temperature of −14° F. in the evaporator and it is assumed that there is no pressure drop between the evaporator and absorber, at what temperature must the strong aqua leaving the absorber be maintained to secure a strong aqua of 38 per cent concentration?

8. What pressure of steam must be maintained in the generator heating coils to reduce the strong aqua of Problem 7 to 30 per cent strength, if the temperature of the condensing steam is 9° F. above the temperature of the weak aqua leaving the generator, and the temperature of the saturated ammonia in the condenser is 15° F. above the temperature of the condenser water, which is 65° F.?

9. How many pounds of strong aqua must be pumped per pound of ammonia removed from the evaporator when the pressure in the absorber is 25 lbs. abs.; the temperature of the strong aqua leaving the absorber, 101° F.; the pressure in the generator, 185 lbs. abs.; the temperature of the weak aqua leaving the generator, 260° F.?

10. How many pounds of 26° F. Beaumé aqua ammonia must be added to 1000 pounds of 16° F. Beaumé aqua ammonia, in order to bring the strength of the resulting solution up to 20° F. Beaumé?

CHAPTER VI.

THE AMMONIA COMPRESSION SYSTEM.

Horizontal Double-Acting Compressors.—It has been impossible to determine the actual ratio between the number of the installations of horizontal and vertical machines, but the data on this subject seems to indicate that the major portion of the machines which are installed at present are of the vertical single-acting type. Since the horizontal double-acting type of compressor has been used extensively in the past, the description of this type will be given first.

Probably the strongest point in favor of the selection of the horizontal type of machine is the accessibility of the working parts. It is apparent that since the working parts are so near the floor they are always readily accessible and under the care of the operator. Thus the construction gives a very compact arrangement, which makes the machine easily overhauled, repaired and operated.

A second point in favor of the horizontal type is that the machine gives good service under any condition of operation that may occur. It operates in an efficient manner when working under either wet or dry compression. Thus, it is a dependable unit, while on the other hand the vertical type may give trouble when handling wet vapor.

The horizontal machine requires nearly the same floor space as the vertical type, but the head room required is about one-half that occupied by the vertical machine. The horizontal machine has fewer moving parts and smaller amount of rubbing surfaces. The friction of the machine is, therefore, less; the machine may have a greater mechanical efficiency; and there are less liabilities for accidents.

The center of gravity and lines of action of stresses in the frame are very near the floor line. A firm foundation and connection between the frame and foundation may be made. This produces a durable construction that makes the operation of the machine noiseless and without vibrations.

Since the horizontal machine has a smaller amount of rubbing surfaces than the vertical type, it requires less lubricating oil, and can be kept in a much cleaner condition.

The metal in the frame is distributed in such a manner so as to give the greatest strength with the least amount of metal. The interior ribbing is so designed as to produce rigidity. At the same time, the metal is distributed in a manner to reduce casting stresses, and to give a broad and continuous contact with the foundation. The frames are usually made from a special mixture of close grain cast iron. The frames are machined by special machines and methods that insure perfect alignment of parts. Frames are cast in one piece until the larger sizes are reached.

The compressor cylinder or bushing is made of a special mixture of iron and steel and is cast separate from the frame. It is machined on the ends on the outside, and rough turned on the inside and ends. It is then tested at a high pressure, after which it is pressed into a frame and the machining and finishing are done. This insures perfect alignment and fit between the cylinder and cylinder heads. The bushing is held by the grooves in the cylinder head. The compression stresses are therefore carried by the frame. This tends to keep the machine in alignment. The annular space between the bushing and frame is used as a water jacket to keep the machine from becoming overheated.

The compressor heads are cast from a special mixture of iron and steel and are designed for maximum strength. The inner surfaces of the heads are finished to a spherical form, generally giving strength and large valve area. The heads are held to the frame by heavy studs, with gaskets between the head and the bushing that produce ammonia and water tight joints. The valve chambers are cast in the heads and are bored radially. Greater number of valves are used on the larger machines.

The piston is generally a light and strong casting of semi-steel. It is generally cast in one piece and is hollow. The faces of the piston, of course, are turned to fit the spherical surface of the heads. They are pressed on the piston rod and held by riveting or by a nut and riveting. They contain grooves into which are fitted a number of packing rings. A built-up piston is sometimes used, which consists of spider, spring rings, and bull rings. Flat faced pistons are also used.

The stuffing box is a long and deep compartment extending around the piston rod. The function of the box is to prevent the leakage of the ammonia. It must remain cool to operate with a minimum amount of friction so as not to score the rod.

The main stuffing box contains two sets of packing separated by an oil lantern. The number of packing rings varies with the type of packing and the size of the machine. The oil lantern is a casting which provides a clearance space all around the rod. The outer end

of the packing is retained by means of a stuffing box gland, which is held by screwing or bolting to the neck of the cylinder head. The oil lantern is connected to the suction manifold through a valve. A connection is made at the same point to the liquid line for expansion into the suction manifold to help hold the temperature at a lower point, if so desired. The oil lantern compartment is connected to an oil pump for supplying a means of lubrication. There is an outer set of packing, which is held in the stuffing box gland by a nut that screws into the gland. Various methods are used to advance the stuffing box gland evenly into the packing compartment, such as the use of gears and screws.

The valves on compressors may be the balanced poppet type. The stationary parts are made from cast iron and the working parts from steel. The valve cages are held in the valve ports by means of a retaining nut that screws into the cylinder head and also holds the cage on a ground seat tightly. Suction valves have stems that prevent them from falling into the cylinder if they should break. Valves are balanced by a spring of the proper strength. They are prevented from slamming at the end of the opening by means of a cushion spring or by a gas cushion chamber.

Plate valves are used also. These will be described later.

The crosshead is of a box construction. It generally has curved movable shoes on the top and bottom. The shoes are adjustable along the inclined guides by horizontal screws. The shoes are lined with babbitt. The box is made of semi-steel, the shoes of cast-iron. The pin is made of steel and fitted into the crosshead on a ground taper fit, being secured by a nut. The piston rod screws into the crosshead and is secured by means of a heavy lock nut.

The connecting rods are made of forged steel and solid from end to end. Recesses for brasses are drilled and milled out, giving a strong rod. Accuracy of adjustment of brasses is accomplished by means of wedges and screws. The crosshead brasses are made of bronze, while the crank-end brasses are lined with babbitt, generally. The brasses are centered by means of flanges or grooves.

The advantage of having a perfectly straight crankshaft is obvious; it does not produce uneven wear on the bearings. The shaft is machined from a steel forging. The crank or crank disc is a casting, and after being machined is forced on the shaft by hydraulic pressure or by shrinking. The crank pins are forged steel and are forced into the crank by a heavy pressure and then riveted over.

The main bearing is usually constructed with floating or loose boxes on the lower and upper sides. This is known as the floating quarter box type. The sides of the bearing are adjustable quarter boxes, which are adjusted by means of wedges and screws. The

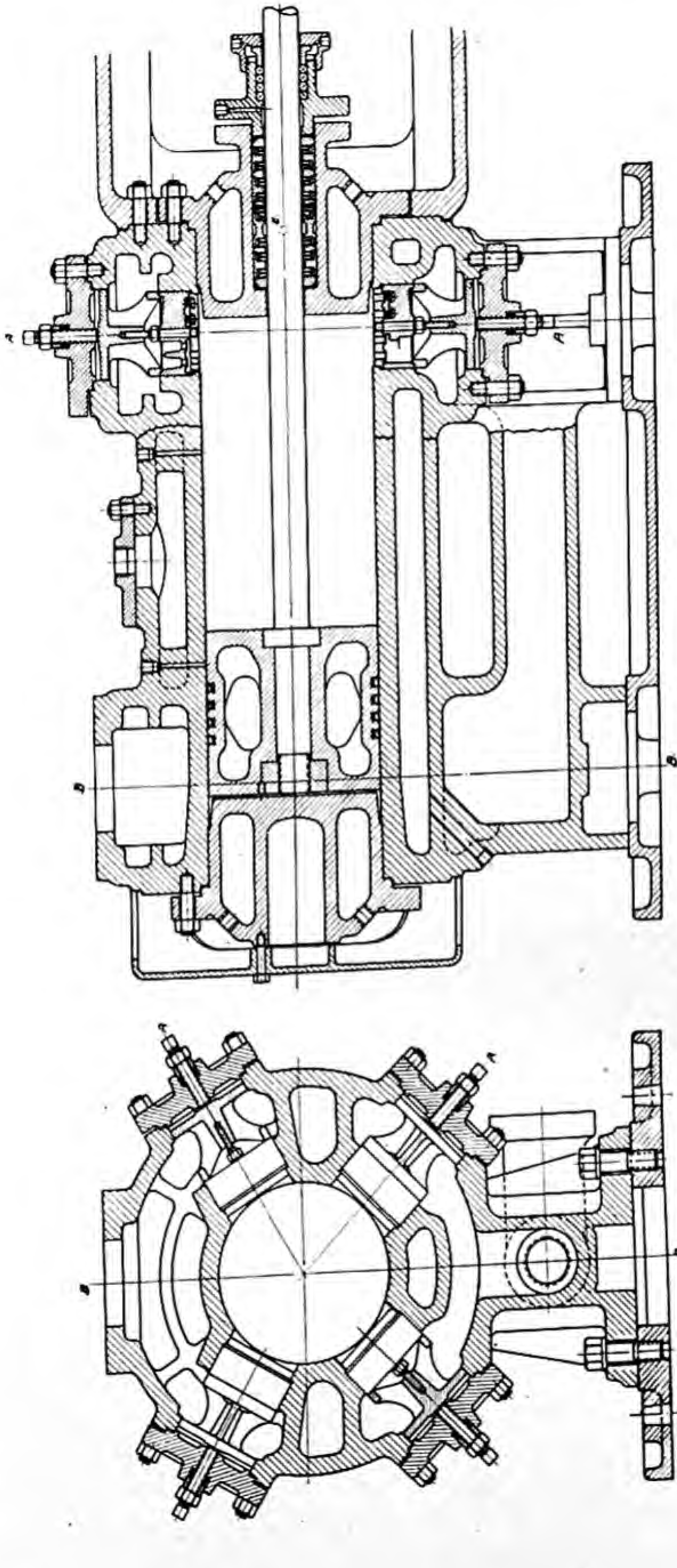


Fig. 55.—Frick Medium Speed Ammonia Compressor.

bearings are lined with babbitt metal. The adjusting screws pass up through the bearing cap and are held in position by lock nuts. The boxes are held in position by flanges or collars. The bearing cap is secured to the frame by means of heavy studs.

The outboard bearings have the same design as the main bearings. The bearings are placed on a capstone and are secured by means of foundation bolts passing through an adjustable sole plate. Another form is that of the pedestal type. This type sets upon a small capstone and is secured by foundation bolts.

Liberal provision should always be made for the thorough and systematic lubrication of all rubbing surfaces. A hand oil pump is generally supplied for the stuffing box; adjustable sight feed oilers, for crosshead guides and main bearings and stuffing box gland; a wiper oiler and cup, for crosshead pin; centrifugal oiler of pendulum type, for crankpin. A more efficient system is secured by installing a central oiling system consisting of pump, filter, storage tank, and connections to supply oil continuously to the rubbing surfaces.

Fig. 55 shows the construction of the Frick medium speed horizontal double-acting ammonia compressor. This compressor is representative of the several makes which are on the market at present.

High-Speed Horizontal Ammonia Compressors.—The principal advantage of the high-speed ammonia compressor is the ability for direct connection to electric motors, high-speed steam, gas or oil engines. Direct connection results in a saving of power, due to elimination of belts, chains, and ropes for driving. Another material advantage is the saving of space required for installation. A saving of total cost is thus effected.

The frame is of the rolling mill type. The metal is so distributed as to give maximum rigidity and at the same time have a pleasing appearance. It also allows a broad and continuous contact with the foundation. The frame is cast separate from the cylinder, thus permitting ease in handling and reducing shrinkage stresses in casting. The cylinder casting is secured to the frame by means of heavy studs. The frame is machined in a careful manner to insure perfect alignment. Shoulders and grooves are provided to insure alignment of parts bolted together.

The compressor cylinder is constructed of close grain gray cast-iron or of semi-steel. The valve ports are placed radially around the ends of the cylinder. The cylinder is supported by a sliding support under the cylinder to help carry the weight of the connections, etc. The cylinder heads, of course, carry no valve chambers, as in the case of the slow-speed machine. The heads are made of semi-steel or gray

iron. They may be arranged for water cooling and this is the place where cooling has the most appreciable effect.

The piston is made of gray iron or semi-steel and is cast hollow. It is pressed on a rod against a shoulder and is secured by means of a heavy nut, which is prevented from turning by pouring in soft metal around the nut after assembling or by other means. Spring rings are put in grooves on the piston.

The valves are usually of the ring plate or feather type; they are made of a special steel by a special heat-treatment method. They are thin and light and are ground to a smooth surface. The ring plate valves are held on the seat by means of a light spring. The feather valve, of course, requires no springs. The valves operate noiselessly at high speed and permit of a large valve area. The plate valves introduce more clearance than is found in a slow-speed machine, but this is not objectionable since clearance helps to produce smooth running at the higher speeds.

The high-speed machine necessarily requires a good system of lubricating the bearing surfaces. An automatic central oiling system is generally used. This consists of an oil filter, storage tank, water separator, oil pump, connections, and sight feed valves. All moving parts are protected by planished steel guards, which are removable. By this means, ample quantities of cool, clean oil, are supplied to the bearings. The oil is used over and over again, and the system is generally automatic and reliable in service.

Fig. 56 shows the general construction of a Vilter high-speed double-acting compressor.

Fig. 57 shows the construction of the Vilter plate valve for both suction and discharge. The valve is of the plate type and has met with immediate and unvarying success. It is made of thin, hardened steel and is ground down to surface. Being of inappreciable weight its inertia is small even at high speed and in consequence noiseless operation may be obtained at high rotative speeds. On account of this peculiar construction, which allows large valve area, it is possible to keep the gas velocity through the ports down to a nominal velocity even at high speeds. The eight suction and eight discharge valves in each cylinder are identical in construction (but of different sizes) and are only dissimilar in the manner of being placed into the valve chambers.

The plate valve shown in Fig. 57, is so designed as to be absolutely safe. It can be seen from the illustrations that the valve cannot be displaced in any possible manner. The valve cage is held in position in the valve chamber by a special steel stud, which is adjusted in place by means of screw threads cut in the semi-steel cover plate, made tight by means of a special nut.

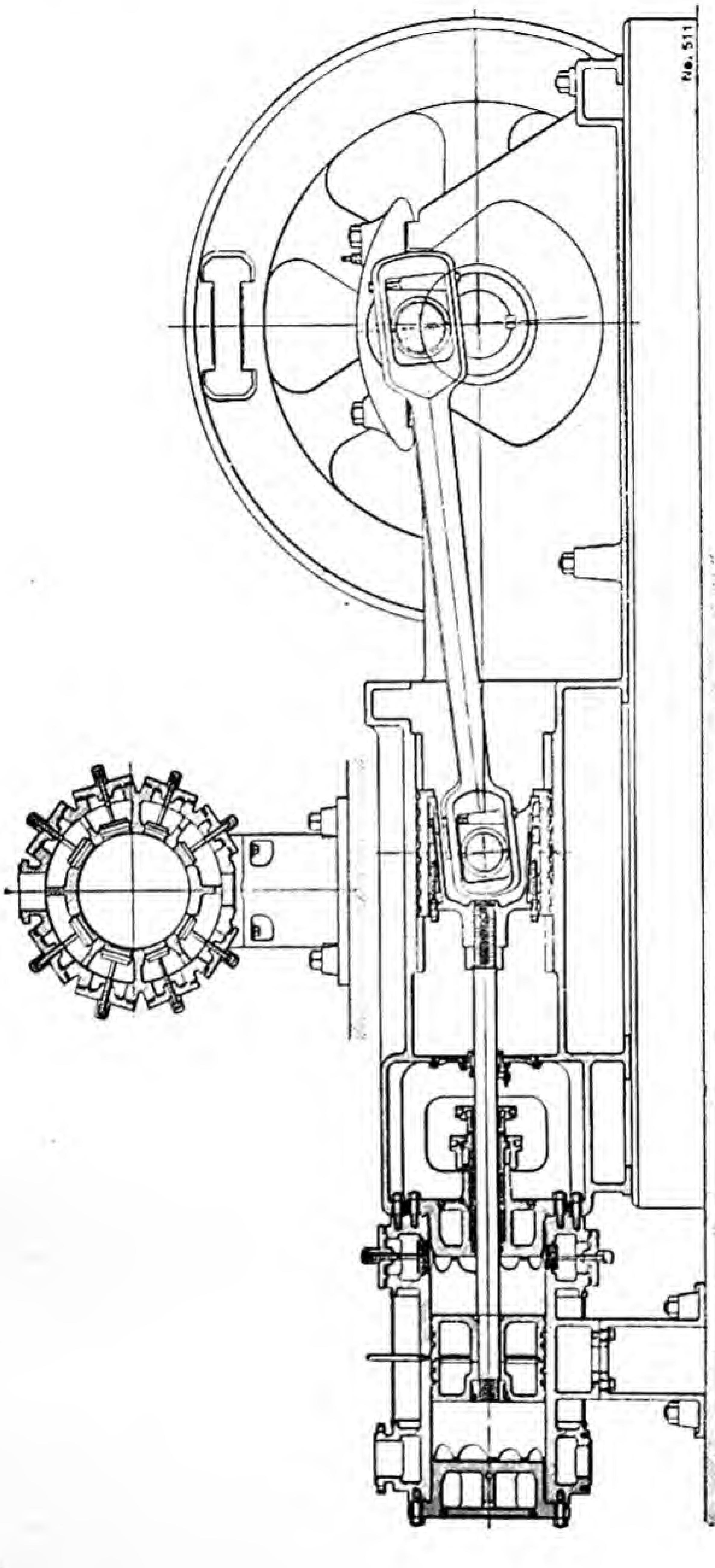


Fig. 56.—Vilter High Speed Horizontal Compressor.

The design of some of the horizontal double-acting ammonia compressor cylinders makes no attempt to jacket the entire cylinder and usually since only the discharge portion of the cylinder is heated during compression, the jacket is applied only to this section and the cylinder heads. Further, it has been found that more important than restricted valve area, is port area, due undoubtedly to the building up of a large volume of gas in the parts less susceptible to inertia effects due to interrupted gas flow and, therefore, high-speed compressor cylinders have ample port area. Fig. 58 shows the construction of a Carbondale ammmonia compressor cylinder to illustrate this statement.

Multiple Effect Compression.—Consists of operating under two different suction pressure conditions with a common discharge pressure and is made possible by having ports placed at a suitable point in the cylinder barred so that they are uncovered by the piston when nearing the end of its stroke, those ports communicating with a space or belt surrounding the cylinder. In horizontal double-acting compressor, these ports will, of course, be in the middle of the cylinder barrel necessitating the use of a long piston, the principle of construction being very similar to the cylinder of a Unaflow steam engine.

The piston during its suction stroke first of all draws in gas at the low pressure suction condition and upon uncovering the multiple effect ports, gas at the multiple effect suction pressure condition flashes into the cylinder, the amount entering being predetermined by the effect that this high pressure suction gas has in virtually compressing the low pressure gas to this condition. From this point, the gas is then compressed in the cylinder in the same manner as in a simple compression machine, therefore, the indicator card is very similar to the indicator card for a simple compression machine operating between conditions similar to the multiple effect suction condition and the final discharge pressure condition with the rectangular port added below involved by the multiple effect compression. Therefore, it can be readily seen that to calculate the hp. for a machine operating with this type of compression it roughly means determining the mean effective pressure for an indicator card operating between high suction pressure condition and discharge, and adding to that a mean effective pressure equal in amount to the difference between the two suction pressure conditions.

Multiple effect compression was originally introduced for use in the tropics on CO_2 compressors where the available cooling water was always at a higher temperature than the critical temperature of this particular refrigerant. Therefore, the problem of liquefying the gas was rather difficult. In order to get the greatest amount of refrig-

eration, we all know that it is necessary to take advantage of the latent heat of the refrigerant in changing from the liquid to the gaseous state at the low temperature condition. By introducing the multiple effect suction condition a small amount of refrigeration was gained between high pressure condition and the multiple effect suction condition, the gas up to this point not having been liquefied, this refrigeration in turn being used to liquefy the gas before entering the evaporator thereby materially increasing the overall gain in refrigeration for the complete unit.

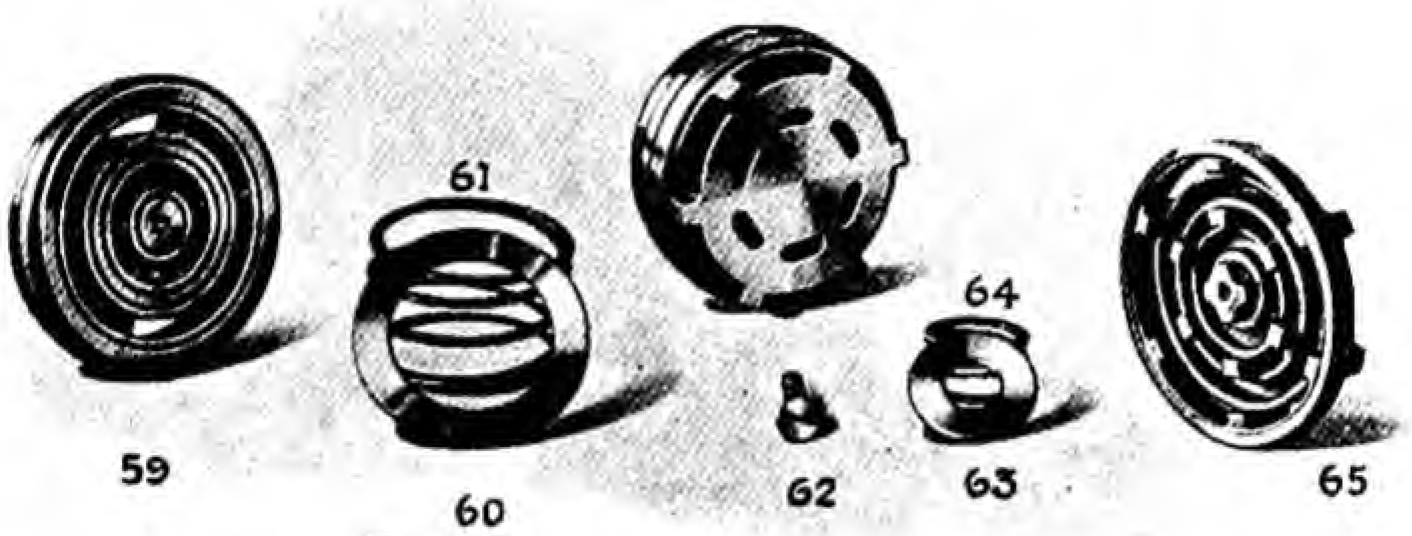


Fig. 57.—The Vilter Plate Valve for Suction and Discharge.

The principal advantage in multiple effect compression so far as ammonia refrigeration is concerned is mainly evident in small ice cream manufacturing plants where two different conditions of low temperature are necessary, one for freezing the ice cream and another for the hardening process. In this way, by adopting the multiple effect compression, one machine can take care of the complete plant where two were otherwise necessary. We have, of course, instances where simple compression machines are operated with two different suction conditions on either end of the compressor, and in some cases where three different temperatures are required, double suction connections have been furnished and multiple effect operation.

A construction which has proved very popular on horizontal compressors is that of the "Doubleseal" stuffing-box (patented) as illustrated in Fig. 59.

The characteristic effect of a "slop-over" which would tend to increase packing leakages due to contraction of the metallic rod will be overcome in this design because the outer box seals a portion of the rod which does not enter the cylinder and is therefore not affected by the temperature change.

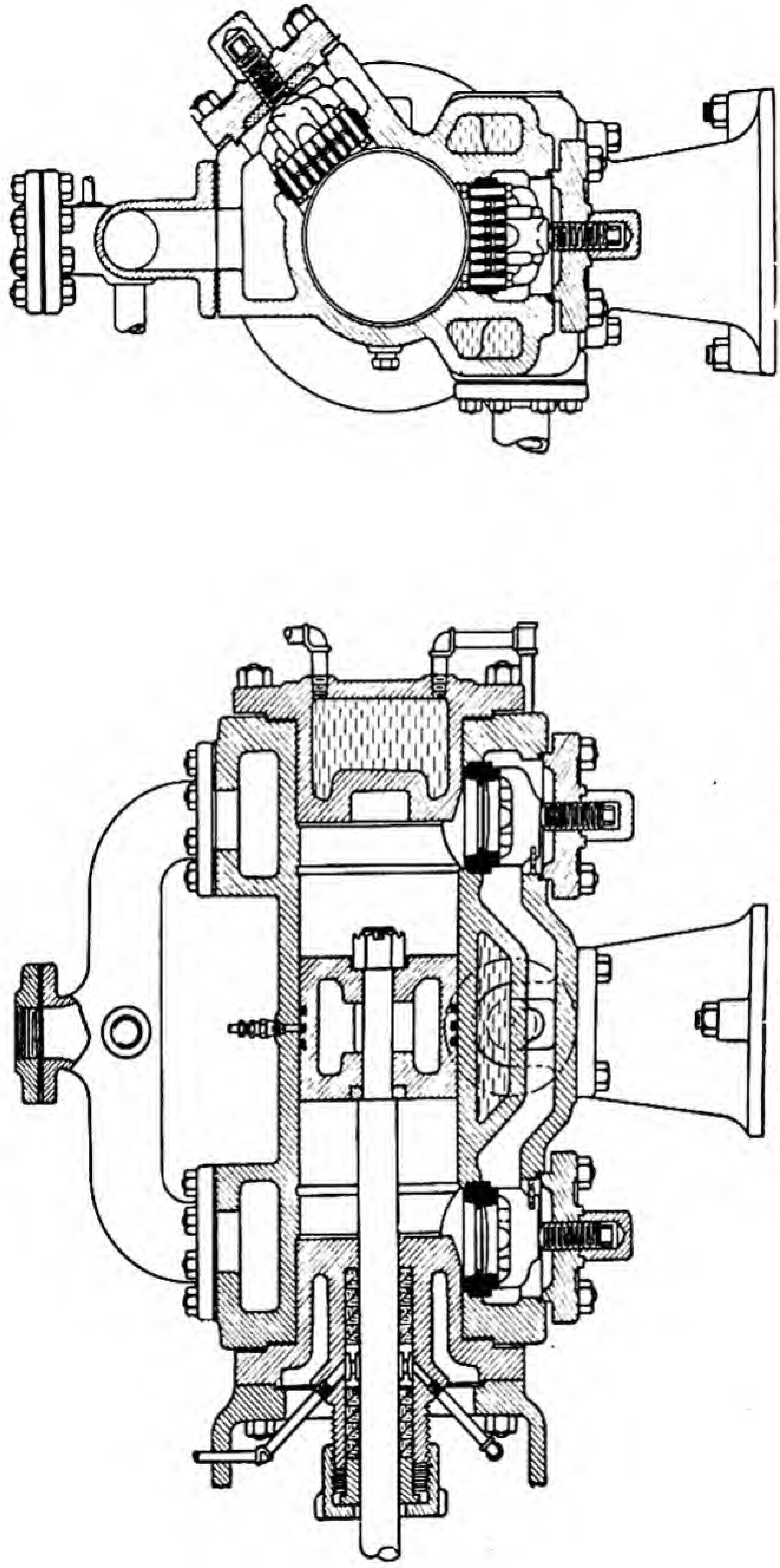


Fig. 58.—Carbondale High Speed Horizontal Compressor.

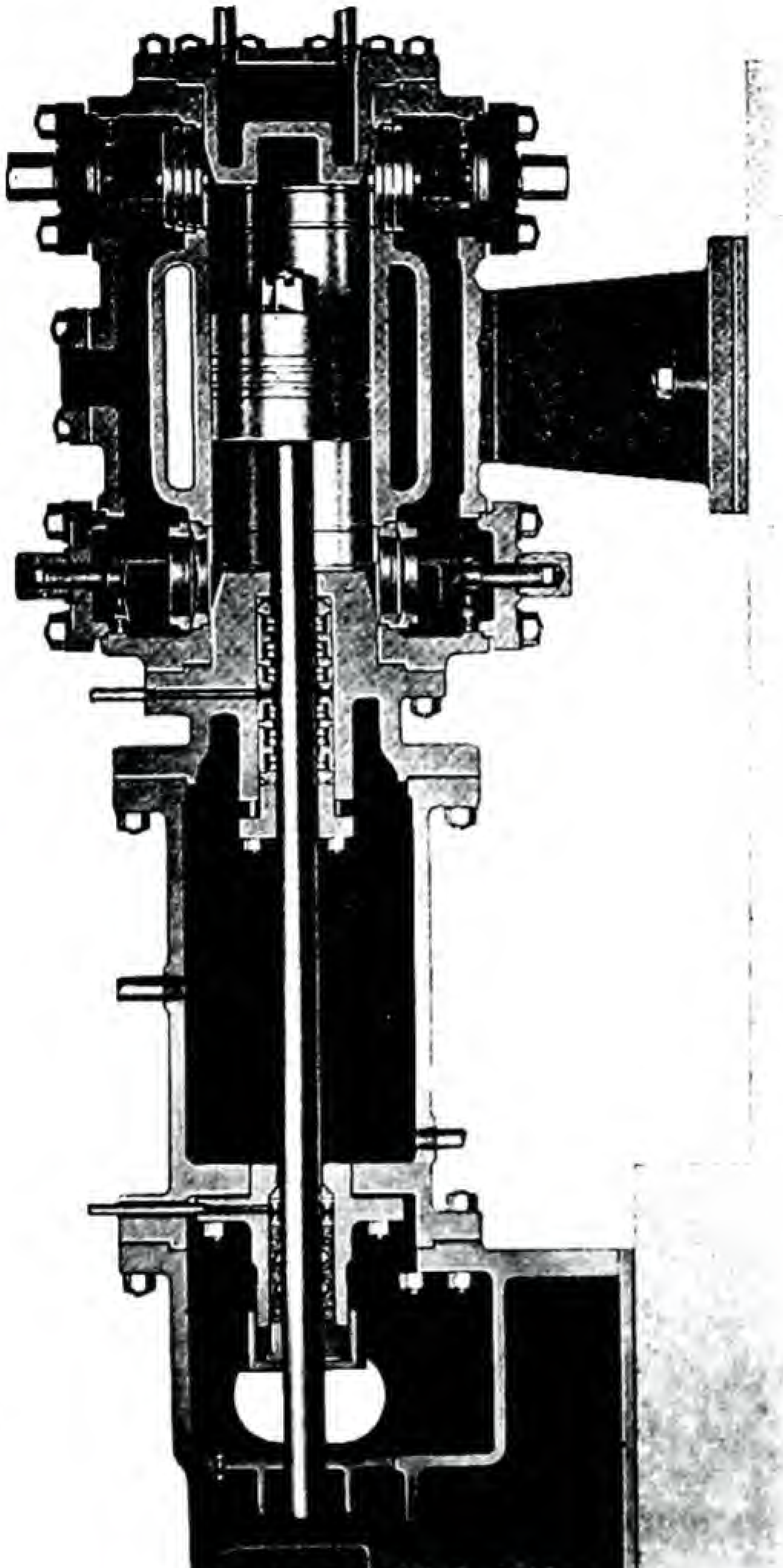


Fig. 59.—Sectional View of a Worthington Feather Valve Ammonia Compressor Showing Double-Seal Stuffing Box.

The "Doubleseal" stuffing-box has a double advantage when installed in any compressor where suction pressure is below that of the atmosphere since connection of the chamber to some higher source of pressure eliminates entirely the possibility of drawing air into the system. Under ordinary conditions of operation the "Doubleseal" chamber is connected to the suction of the compressor.

The inner or main stuffing-box is fitted with either metallic or soft packing while the outer box is equipped with soft packing. Both boxes are lubricated with a separate feed to the mechanical lubricator.

Low Temperature Operation.—In oil refinery practice there has been a heavy demand for very cold test lubricating oil. Many ammonia compressor plants have been installed for exceedingly low temperature which are used to freeze the wax out of the oil, so that it will not congeal at sub-zero temperatures.

These plants usually operate at an ammonia evaporating temperature of minus 65° F., which is accomplished by the use of a two-stage compressor with a separate reciprocating booster taking the gas at almost 5 lbs. absolute and compresses it to almost 15 lbs. absolute to the suction of the two-stage machine. Heat exchangers are used on the booster suction to cool the anhydrous and at at the same time superheat the suction so that this extremely cold gas need not be handled in the compressor. In general, the operation can be considered as three stages of compression with adequate cooling between stages.

Dry Ice Compressors.—Recently especial attention has been given to the manufacture of compressors for liquefying CO_2. It is of course known that CO_2 gas is first liquefied by compression to 900 to 1200 lbs. and after cooling the liquid gas as much as possible it is expanded through an expansion valve into the snow chambers from which the snow is pressed into cakes. Only a part of the gas freezes into snow, varying between 20 and 40 per cent, depending upon the process, the rest of the gas being returned to additional compressors for recompression and re-liquefaction. This process is then repeated.

Dry-Ice is now being used by most of the ice cream companies for delivery of ice cream, not only for local distribution, but intercity distribution. It is also being applied to the distribution of meats and other commodities.

The condition of operation of the cycle dictates three stages of compression for dry ice compressors.

Two-Stage Duplex Compressors.—Two-stage duplex compressors are similar in construction to the single-stage compressors. Duplex horizontal machines for small capacities, arranged for either long belt-drive or provided with idler for short connection to motor, are

built in either single- or two-stage types, and range in capacity from 50 to 100 tons.

The duplex arrangement also lends itself admirably to two-staging with the increased efficiency due to that construction.

The capacities of such machines vary from 70 to 700 tons in single units and are usually driven by the synchronous type of electric motor. This type of compressor is coming into general use in the majority of large plants because of the economies of both the motor and the compressor, particularly when the latter is equipped with clearance pockets for capacity control.

Intercooling.—The utmost economy in two-stage compression is secured only with the proper type of intercooling. There are three practical methods of obtaining an intercooling effect, viz:

(1) Water Intercoolers.
(2) Liquid Ammonia Coolers.
(3) Liquid Ammonia Injection.

Water Intercoolers.—This type of intercooler is similar in construction to the multipass tubular condenser, and can be placed in either vertical or horizontal position, adjacent to the compressor cylinders. Or an intercooler of the vertical single-pass type can advantageously be used on larger units.

The economy of this intercooler is influenced by the temperature of the water available. The colder the water used, the greater the benefit derived from this type of intercooler.

Liquid Ammonia Coolers.—Where added results are desired, this type of intercooling will give greater efficiency. It is constructed somewhat similar to an accumulator, consisting of a steel shell in which liquid ammonia may be expanded and containing a liquid precooling coil through which all the ammonia used in the plant flows. The liquid ammonia is cooled from the condenser conditions to a temperature corresponding to the pressure in the intercooler. To effect this it is necessary to expand ammonia in the intercooler, this expansion also cools the gas passing from the low- to the high-pressure cylinder, removing the excess heat and so reducing the volume of the gas that the high-pressure cylinder can perform more work. The use of this cooler will effect a saving in power of from five to fifteen per cent.

Liquid Ammonia Injection.—The third form of intercooling, and the lowest in first cost, about equalling the water intercooler in power saving, is accomplished by injecting a small quantity of ammonia liquid into the suction line of the high-pressure cylinder, thus cooling the gas from the low-pressure cylinder discharge temperature to the

vaporization temperature of liquid ammonia at intermediate pressure. The liquid ammonia may be injected automatically by means of a no-freeze-back control valve.

In special cases, a combination of water and ammonia liquid intercooling might not only be feasible but practical in view of the added economy obtained.

Clearance Pockets for Ammonia Compressors.—One of the recent developments in ice making and refrigerating machinery is use of clearance devices or pockets on ammonia compressors. The main purpose of such clearance pockets is to permit the operator to vary the capacity of the machine according to the demand of refrigeration. Due to the fact that many of the modern compressors are driven by constant speed motors, the problem of varying the capacity of the machine to suit the load presented itself.

This problem has been solved practically by fitting a special head on the head-end of the compressor cylinders. In these special heads are a number of clearance spaces or pockets. Usually, three such pockets are used on each cylinder head. Each pocket is connected to the inside of the cylinder by means of a special built-in stop valve, so that the capacity of the head-end of the cylinder may be reduced from full capacity to zero capacity in three steps.

The general construction of the clearance heads for ammonia compressors is shown by Fig. 60, in the application of clearance heads to a two-stage compressor. The clearance heads are bolted to compressor frame just as a standard head. The cylindrical extension contains the three pockets and three special valves. The valve stems extend through the end of the head extension in stuffing boxes. Suitable hand wheels are provided for operating the valves.

By opening one of the valves, a certain amount of clearance space is connected to the head-end of the cylinder. When the piston approaches the head-end, the gas is compressed into the clearance space; when the piston recedes, the gas in the clearance expands through the special valve into the cylinder again. The amount of clearance required for such pockets depends upon the relative size of the cylinder and a number of other conditions which are discussed in the following paragraphs.

The amount of clearance to be introduced into the cylinder in order to obtain a given capacity reduction depends upon the relative suction and discharge pressures. In speaking of the amount of clearance, it is commonly expressed as a percentage of the stroke—volume of the cylinder. By means of the laws of gas compression, as given in most books pertaining to thermodynamics, it is possible to calculate the necessary clearances for various capacity reductions. The volumetric

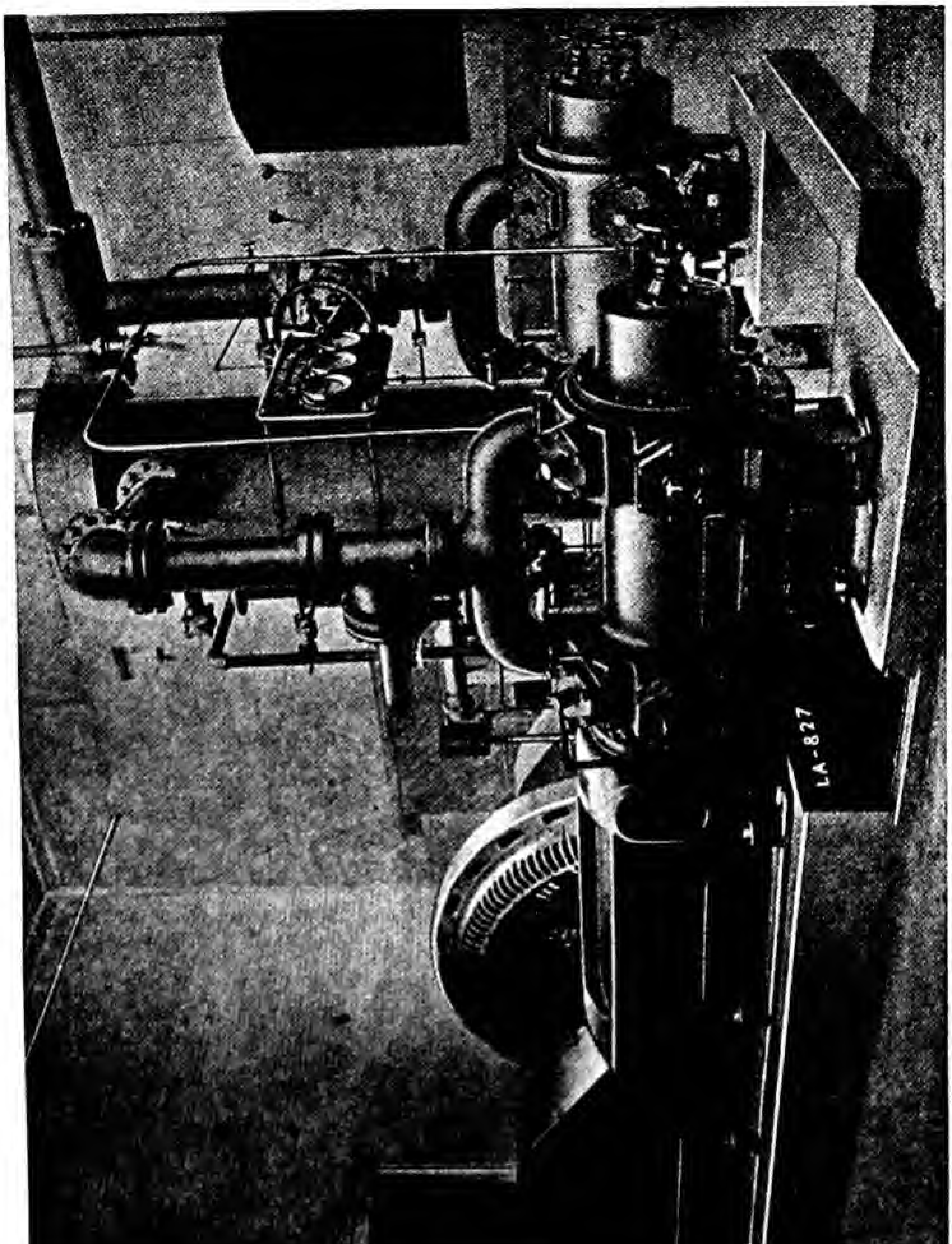


Fig. 60.—Illustrating Appearance of Clearance Pockets

or displacement efficiency due to clearance is shown by the thermodynamic formula in Chapter IV.

The amount of clearance to be used in order to reduce the capacity to zero may be determined also by the foregoing formula.

It is possible to derive a formula, from the above equation, which will express the clearance requirements for zero capacities. The equations are given as follows:

$$E_c = 1 + C - C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

Let $E_c = 0$

C_1 = clearance for zero capacity

$$0 = 1 + C_1 - C_1\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

Therefore

$$C_1 = 1 \div \left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \qquad \text{(2)}$$

Using this formula the clearance requirements for zero capacity at standard conditions may be found as follows:

$$C_1 = 1 \div \left[\left(\frac{169.2}{34.27}\right)^{\frac{1}{1.28}} - 1\right]$$

$C_1 = 0.403 = 40.3$ per cent.

Consequently, a clearance of 40.3 will reduce the capacity to zero for standard pressures.

Formula 2 may be simplified in the following manner:

$$\text{Let } K = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$\text{Then } C_1 = \frac{1}{K-1}$$

The following tabulation gives the values of K for various ratios of compression:

$p_2 \div p_1$	K	$p_2 \div p_1$	K
3	2.36	10	6.04
4	2.95	11	6.51
5	3.51	12	6.96
6	4.05	13	7.41
7	4.57	14	8.85
8	5.07	15	8.29
9	5.56		

The effect of clearance may be shown graphically on the pressure-volume or indicator diagram. In Fig. 61, the diagram AEBF represents the relationship of pressures and volumes, when the compressor cylinder has no clearance. Line AE represents the compression of the gas. Line EB represents the expulsion of the gas from the cylinder. Line FA represents the suction stroke of the compressor. Theoretically, it will be observed that the capacity is equal to the displacement of the piston. Of course under practical conditions there are unavoidable losses which must be allowed for. It is further observed that the area of diagram AEBF determines the amount of work required for the compression and expulsion of the gas. Now if clearance is introduced into the cylinder by means of clearance pockets, it is possible to reduce the capacity to zero, as previously indicated. Referring to Fig. 61, line BC represents a clearance volume of 40.3 per cent. In other words, BC ÷ BD = 0.403. The compression of the gas under these conditions is represented by the compression curve AB. The volume of gas in the cylinder and clearance pockets at the beginning of compression is represented by line DC; the volume of gas in clearance after compression is represented by line BC. The piston travels the entire stroke before the pressure reaches 169 lb. at point B and the piston recedes to the end of the stroke before the pressure is reduced to 34 lb. at point A.

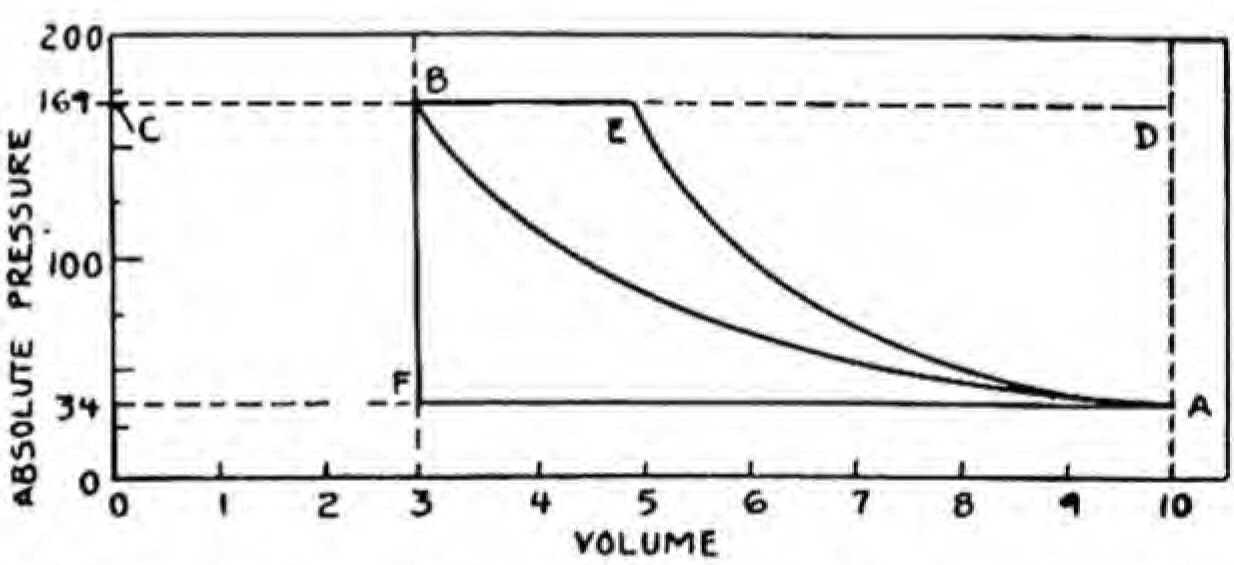


Fig. 61.—Pressure Volume Diagram Showing Action of the Compressor With and Without Clearance Pockets.

Curve AB therefore represents the compression and expansion of the gas in the cylinder and clearance pockets.

As regards work requirements, it will be observed that, since the compression and expansion of the gas follows the curve AB, no work is required, theoretically speaking. Practically some energy is lost through friction and radiation, but the major portion of the energy given to gas during compression is recovered again, during the expansion of the gas.

The details of construction of one type of clearance pocket head are shown by Fig. 62 which is a longitudinal and cross section of ammonia compressor cylinder equipped with such a head. This special head is arranged for bolting on the head of the cylinder as shown. It contains only two pockets or clearance spaces. The clearance spaces are connected to the cylinder by means of suitable ports. The opening of these are controlled by valves, the stems of which extend through stuffing boxes to suitable hand wheels on the outside. To cut in a section of the clearance space the hand wheels on the outside are turned in the proper direction to open the valve in the head. The clearance pockets are cut out of operation by closing the valves.

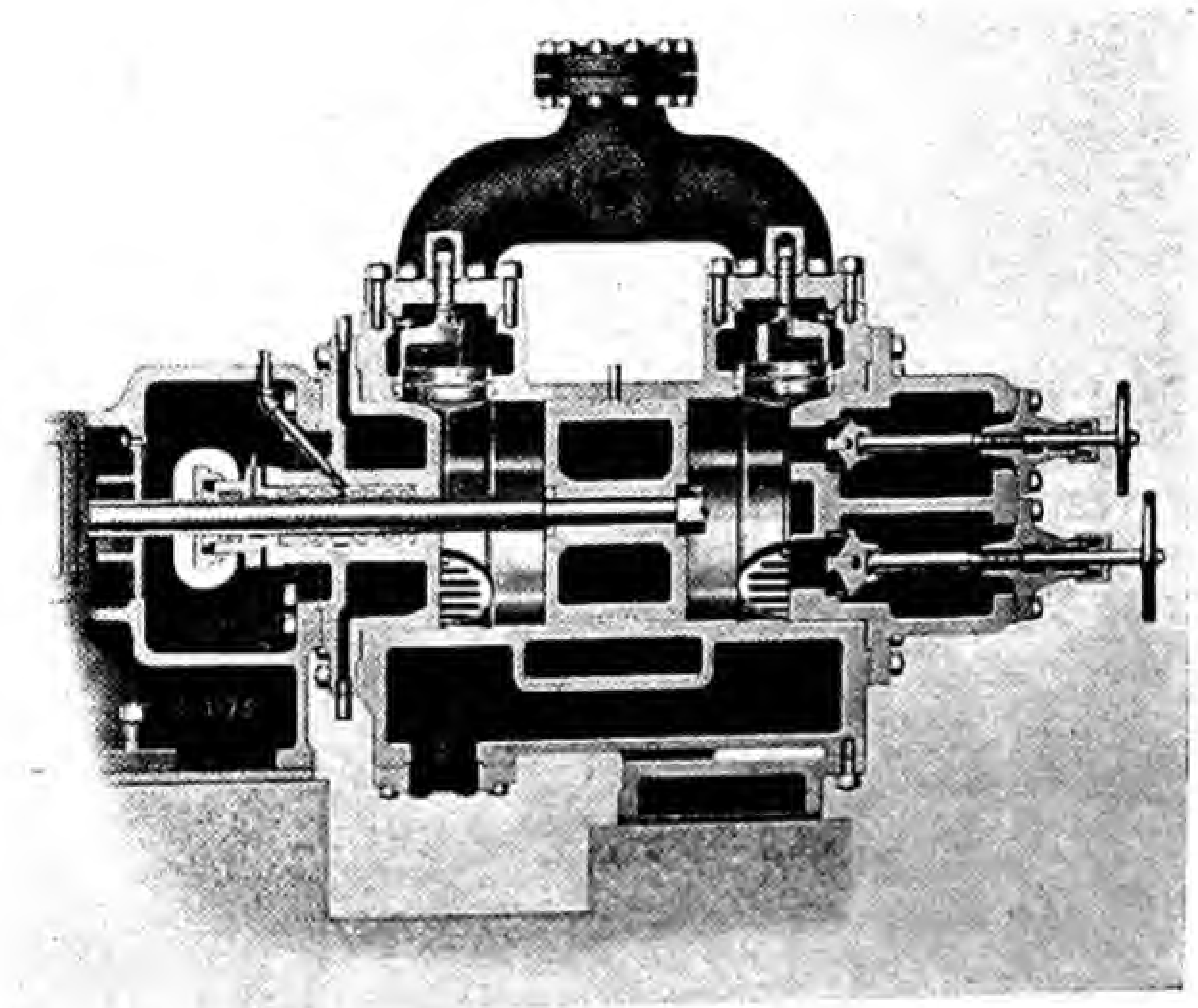


Fig. 62.—Sectional View Showing Construction of Clearance Pockets.

The type of clearance head shown in Fig. 62 makes it possible to reduce the capacity of the head of the cylinder to practically zero in two steps.

The essential advantage in the use of clearance pockets is that obtained by being able to vary the capacity of the compressor in some proportion to the refrigeration demand. As previously indicated, operators of ice and refrigerating plants running under variable loads will appreciate this facility. When the refrigeration load declines, the operator is able to cut in the clearance pockets. The practical result of this operation is that the suction pressure is maintained at a higher average than would otherwise be possible with a constant

displacement compressor. The economic benefit of maintaining a high suction pressure is appreciated when it is remembered that a reduction of one pound in the suction pressure decreases the refrigerating capacity over 3 per cent and increases the power consumption per unit of refrigeration about 2½ per cent. These percentages show why it is desirable to maintain the highest possible suction pressures, yet obtaining the desired temperatures at the points of application of the refrigeration.

Sometimes the following question has been asked regarding clearance pockets: Is it more desirable to install a number of small units without clearance pockets than to install a lesser number of larger units each equipped with clearance pockets?

In the first case, flexibility of operation is obtained by installing several small units, so that machines may be cut in or out of service according to the refrigeration demand. In this method of operation, each unit, while running, would develop its rated capacity, and thus operate at a good efficiency. On the other hand, however, consideration must be given to difference of efficiences of smaller as compared to larger units. Generally, the operating or machine efficiency increases with the relative size of the compressor. Therefore, in the second case, in which larger compressors with clearance pockets are used, the gain in operating efficiency due to larger machine may offset the unavoidable losses of friction and wiredrawing of gas in being compressed into, and expanded out of the clearance pockets.

Other factors to be considered in the economic solution of the question are initial and operating costs of the two classes of machines. The final selection of a given class of machines for a particular plant would be in favor of the type of machine which produces refrigeration at the lowest manufacturing cost, due consideration being given to all the local conditions in the plant.

Quite a number of ammonia compressors are equipped with clearance pockets, at present, in various types of ice and refrigerating plants. The consensus of opinions of operators of such equipment indicates that the clearance pocket is an adjunct to the efficient operation of ammonia compressors.

Vertical Single-Acting Compressor.—The vertical single-acting, false or safety head, ammonia compressor is recognized as an efficient type of compressor. With balanced suction valves of large area, these machines have a high volumetric efficiency and hence a high overall efficiency. The cylinder walls do not tend to wear out-of-round, and the piston rod and stuffing box are subjected to suction pressure only. It has the disadvantage of requiring high head room, and it is more difficult to adjust the working parts.

On the smaller machines, the bed plate and housing are cast in one piece; in the intermediate sizes, the housings are separate, and the bed plates are cast in one piece; and in the larger sizes the bed plates are cast in halves, which are securely bolted and doweled together. There are but two bearings for the crankshaft, the flywheel being placed between them. Accurate alignment of the shaft is secured by pouring the babbitt metal after the compressors and engine have been lined up. Bearings are provided with adjustable quarter boxes.

The compressor cylinder is known as the single-acting, false or safety head type. The gas enters at the bottom of the compressor and passes up through a balanced suction valve in the piston; it is compressed on the upward stroke to the condensing pressure, and passes out from the cylinder through the discharge valve, or valves, in the center of the safety head. The piston and the bottom of the safety head are faced off square, and as they come nearly in contact a very complete discharge of the gas is effected.

The lower part of the compressor on large machines is generally insulated with cork, and the upper part is water-jacketed. The water-jacket casing may be cast with the compressor. The insulated portion and part of the water-jacket are generally covered with steel lagging.

The suction valve is drop-forged steel and of ample area to give minimum lift. A properly designed spring practically floats this valve on its seat so that it opens to admit suction gas immediately at the beginning of the stroke, and remains open throughout the downward stroke so that the suction pressure in the cylinder is nearly the same as in the suction main. The seat and housing of this valve are easily removed from the piston for inspection.

The advantage of the safety head is the security against the breaking of the compressor in case of accidental breaking of the valves or any other part of the machine. In such case, the safety head lifts and allows the obstruction to pass through, or keeps on lifting until it attracts the attention of the engineer, when the machine can be shut down without the destruction of the compressor, which would result from such causes if solid heads were used. Or, in the case of an overcharge of liquid ammonia, the only result is a hammering of the valves and false heads as they raise to allow the liquid to pass.

The discharge valve, of ample proportions to reduce the lift, is made of malleable iron or forged steel. It is held in place by a cage, which is easily removable for inspection. The forged steel seat for the discharge valve is forced into the safety head. Multiple discharge valves are used on the larger sizes and where high speeds are desired.

Ammonia Compressor Design.—Among the advantages of high-speed machinery is readily discernible the reduction in size for a given

output with consequent decrease in manufacturing, building, and foundation costs. A machine properly designed to operate at high speed will do so efficiently and without undue wear to provide overall economy. Further, direct connection to higher speed motors, permitted a lower first cost of the electric drive.

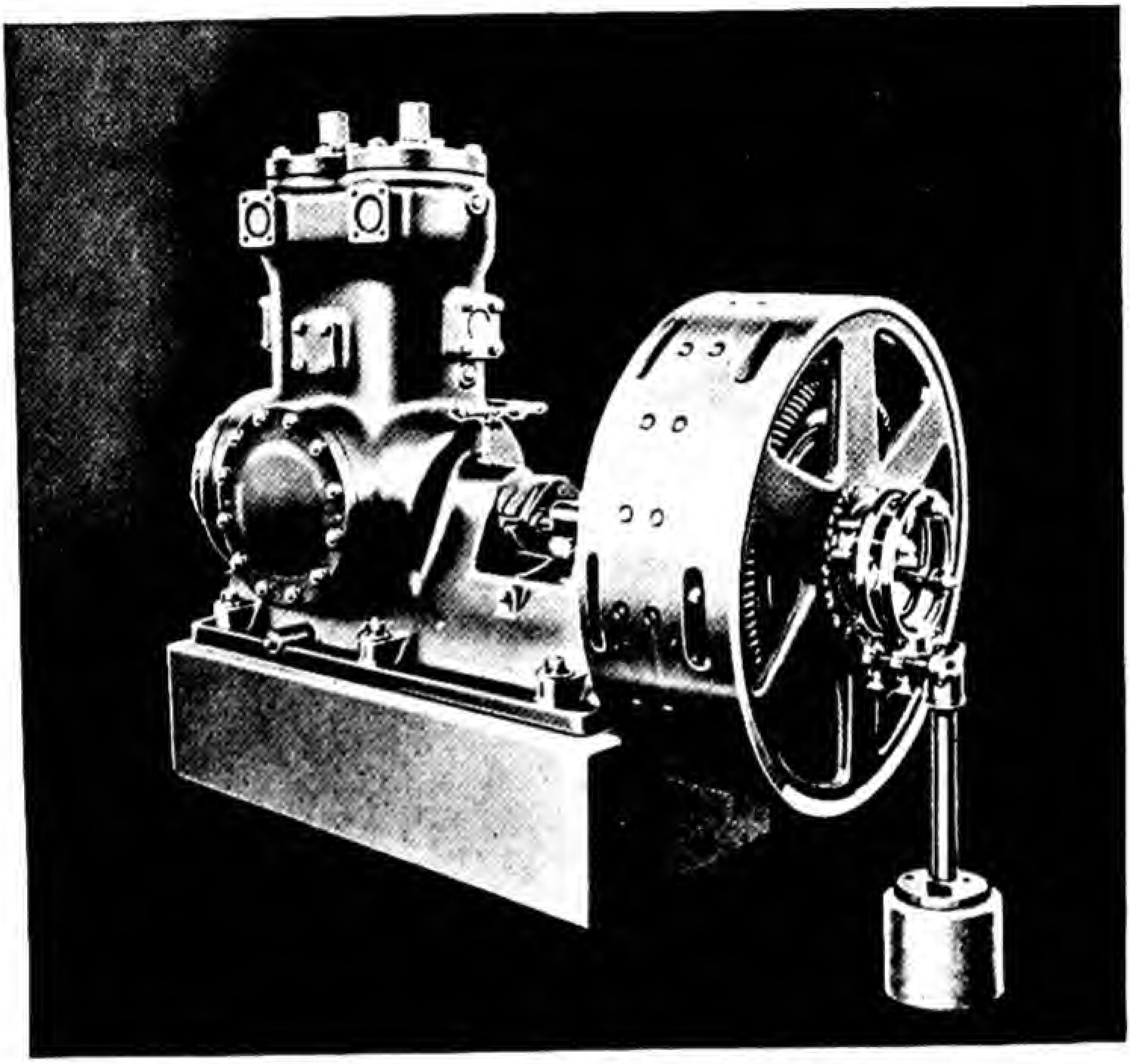


Fig. 63.—Worthington Vertical Single-Acting Compressor with Flywheel Type Motor.

The common parlance in compressor practice is to make the blanket term "high speeds" refer to speeds from 160 to 600 r.p.m. when actually, speeds from 160 to 300 define in general those applicable to the horizontal units, while 300 to 600 apply to the speeds of vertical compressors. These are certainly high speed in comparison with the past custom in ammonia compressor practice.

Vertical Compressors.—The vertical compressor lends itself especially well to the small plant, either for ice making, cold storage, or general refrigeration. Even to supply large loads, some companies have found it advisable to meet their load conditions with the instal-

lation of several small vertical units which can be cut on and off the line when needed.

Vertical refrigerating compressors for the small plant of the enclosed type, are most favorably received in this country for the reason that they lend themselves to high speed, to automatic operation, and to decreased loss of ammonia. This decreased loss of ammonia is due to ease of packing a revolving rod.

Fig. 63 illustrates the type of vertical, single-acting compressor built by Worthington. The cylinder, crank case and base, together with the outer bearing, are cast in one piece of a dense close-grained semi-steel. The casting is sand-blasted and acid-dipped before machining to remove any surface sand.

The cylinder is literally water jacketed to reduce the amount of superheat in the discharge gas.

The cylinder casting with its bearings are machined at one setting assuring absolute alignment which care is especially necessary since this type unit is subjected to very heavy service requirements. The cylinder bore, after being machined, is ground and honed to size, a practice similar to that of the better grade automotive engines. The trunk-type piston is close-grained gray iron and ground to close clearances made possible by honed cylinders. It is fitted with three snap rings above the wrist-pin and one oil scraper ring below, preventing accumulation of oil in cylinder walls that otherwise would finally reach the compression chamber.

The crankshaft is of drop-forged steel, following the highest class automotive practice. The complete shaft is forged in one piece and accurately machined and ground to size.

All bearings are of the die-cast type and are interchangeable. The outer bearing is provided with a ring oiler, dipping into a deep recess in the casting.

The connecting-rod is drop-forged, with a two-part die-casting bearing at the lower end and a removable bushing, steel shell babbitt lined at the upper end. Accuracy in machining, positive alignment of piston and connecting-rod, and interchangeability is assured by the use of jigs. The wrist-pin is hardened and ground to exact size and securely fastened in the piston with locking screws. The connecting-rod is drilled so that positive pressure lubrication is given the wrist-pin.

The change from splash lubrication to full force-feed is in line with modern high-speed development, and is undoubtedly recognized as the safe, practical and efficient means of securing the proper oil supply to a bearing surface. By this means, lubrication is positive, constant and uniform in pressure.

A plunger oil pump directly connected to the compressor shaft extends its suction into the bottom of the crankcase, affording no lift

to the pump itself. Oil is taken from the crankcase below the throw of the cranks and circulated under pressure through the crankshaft which is drilled for this purpose to crank-pins, whence it is taken through drilled connecting-rods to the trunk-pins.

A unique filter is provided on the discharge side of the pump comprising a cartridge of steel plates closely separated, effectively cleaning the oil before it enters the bearings. These laminated filter plates can be cleaned by means of the handle extending outside the case permitting the necessary scraping of filth collection from the filter sheets while the compressor is in operation.

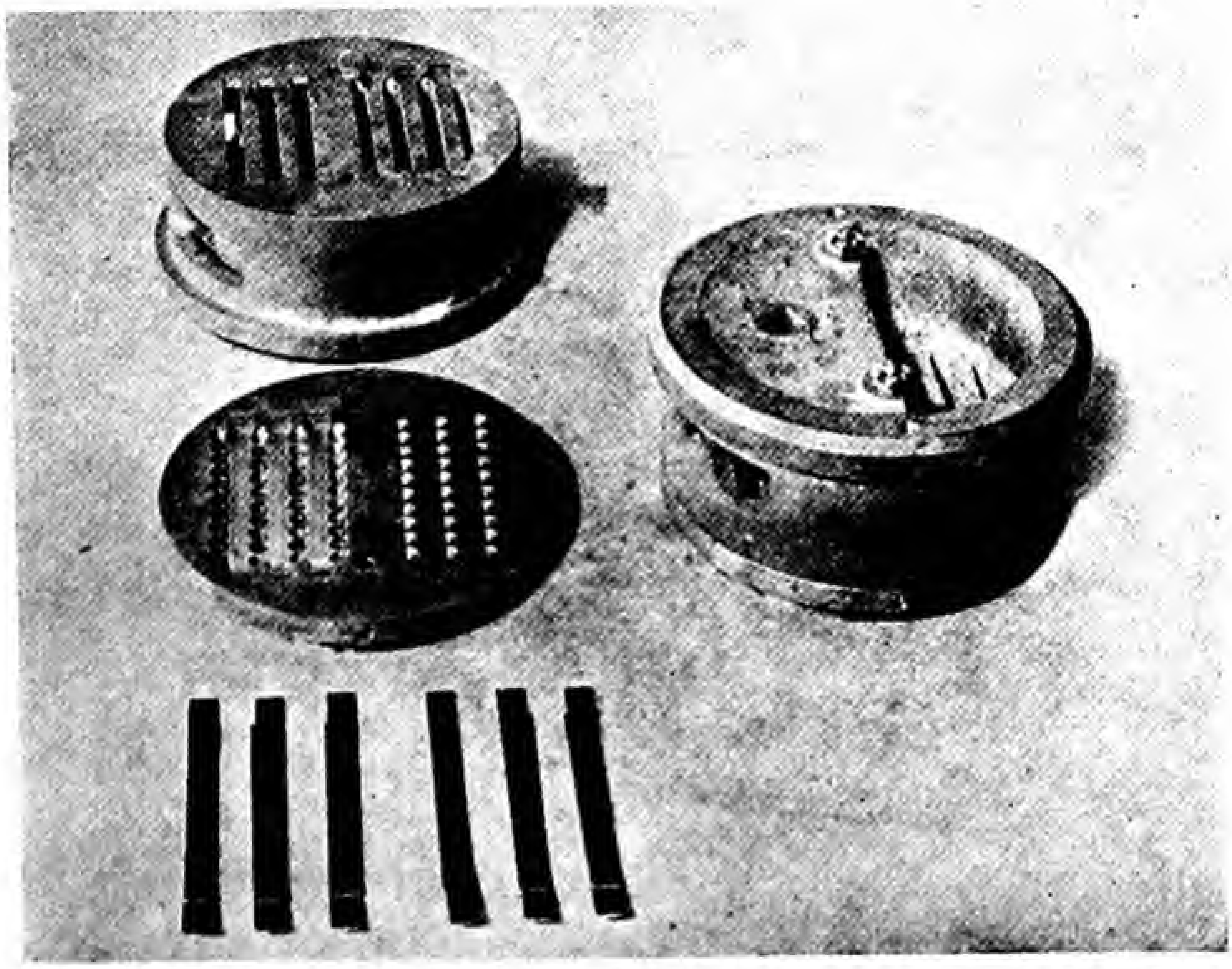


Fig. 64.—Worthington Feather Valve for Vertical Compressors.

Valve Design.—The design of the vertical compressor as built by Worthington is obviously different from other machines, primarily because of valve characteristics. Because the Worthington feather valve, due to its lightness and flexibility, makes practicable a lift considerably greater than is obtained with a valve of any other type and has an area through the lift almost equal to the area through its seat, it has not been found necessary to have a valve in piston as is necessary with the poppet or mushroom-type valve. The lightness of the valve is especially adapted to the higher speeds and therefore the reciprocating motion of the piston is not needed to assist in opening and

closing the suction valve as is necessary with the heavier, greater inertia valves. The valve-in-head, rather than valve-in-piston construction therefore permits of a more compact design, a shorter and

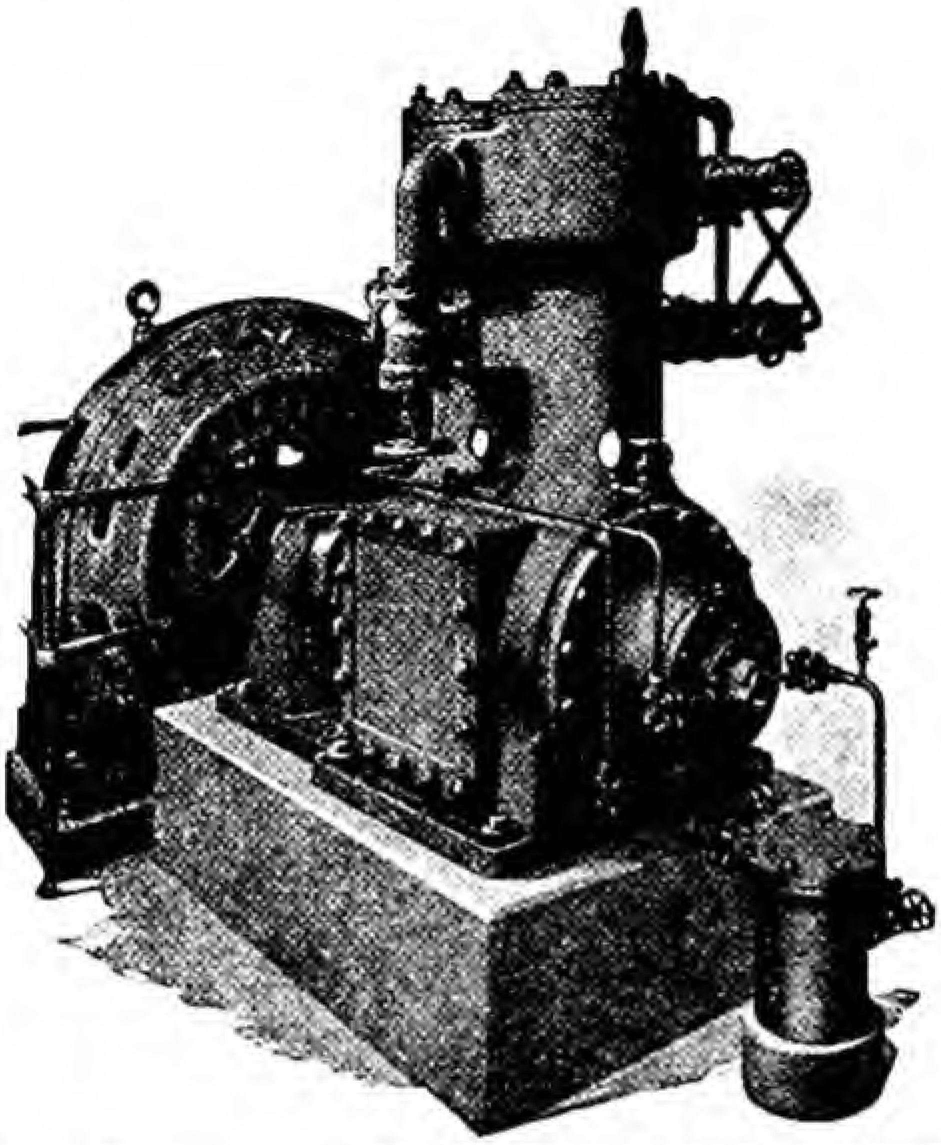


Fig. 65.—Vilter Compressor Direct-Connected to Synchronous Motor.

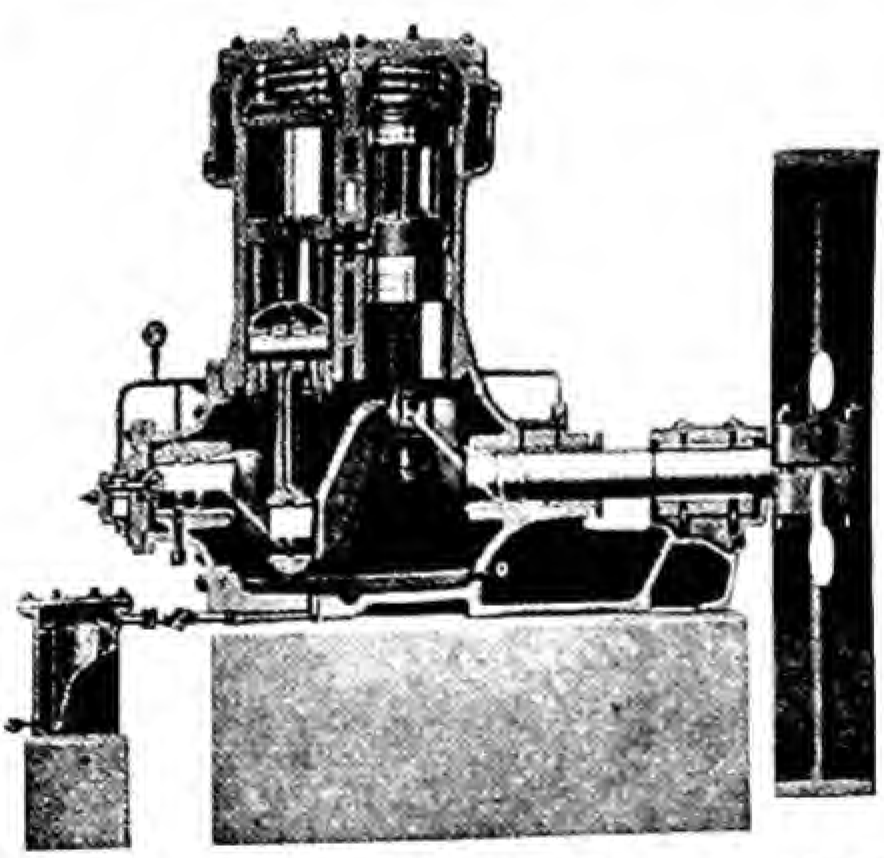


Fig. 66.—Cross-Section Enclosed Vertical Compressor with Force-Feed Lubrication System.

lighter piston giving less weight to reciprocating parts and assures quiet operation due to absence of impact in valves.

Fig. 64 shows features of the valve for vertical units. Both suction and discharge strips are contained in one assembly called the "cage"

which comprises the body and plate with its drilled hole passages. The ground plate is held to the body with ream bolts only as an assistance in assembly, and after installing in the compressor a heavy yoke forces the plate and body together forming a perfect sealed joint between the parts and, at the same time, this force is sufficient to seat the cage against gaskets in the cylinder casting.

Fig. 65 shows a view of a twin-cylinder Vilter ammonia compressor direct-connected to an engine-type synchronous motor.

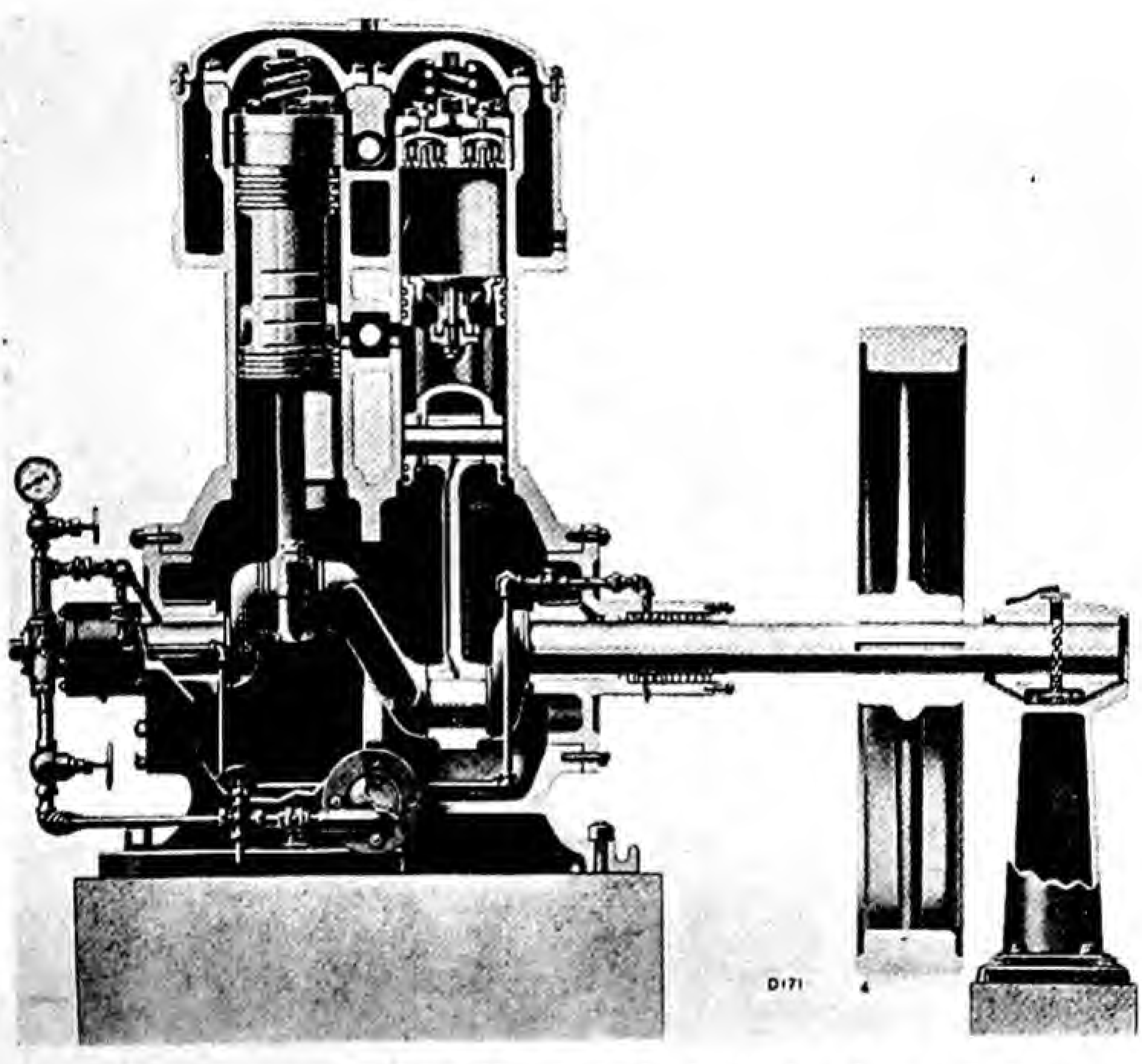


Fig. 67.—Cross Section of Frick Vertical Single-Acting Ammonia Compressor.

A cross-sectional view of an ammonia compressor with its forced feed lubricating system is shown in Fig. 66. The connecting rods are made of cast-steel. Wrist-pins are of carbon manganese steel ground to size. The oiling system consists of a gear-type oil pump, reserve oil tank with strainer, and necessary valves and fittings. Plate-type suction and discharge valves are used. The crankshaft is made of forged high carbon steel. Die-cast babbitt bearings are employed.

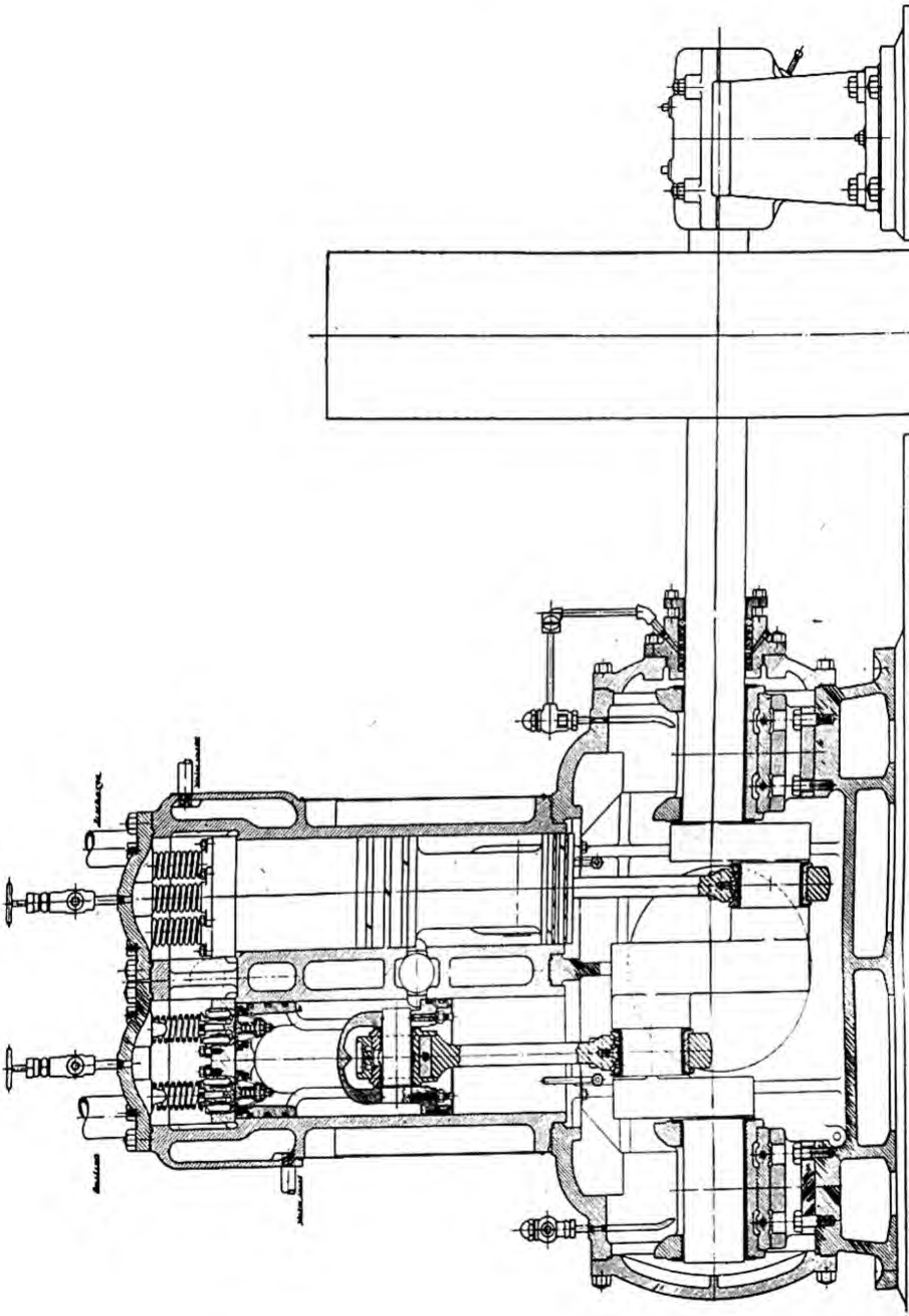


Fig. 68.—York Vertical Enclosed High Speed Compressor.

Fig. 67 shows the construction of the Frick twin-cylinder ammonia compressor of the belted type. It is equipped with a forced-feed lubricating system as shown.

Fig. 68 shows the construction of the high-speed vertical, single-acting enclosed type of compressor.

Fig. 69 illustrates the construction of a unit-type refrigerating machine. Here the compressor, motor, and condenser are mounted on a common structure iron base-plate. The compressor motor is mounted on a steel stand and is connected to the compressor by means of a "V"-belt-drive. The condenser is of the closed shell-and-tube type, connected as shown in the illustration.

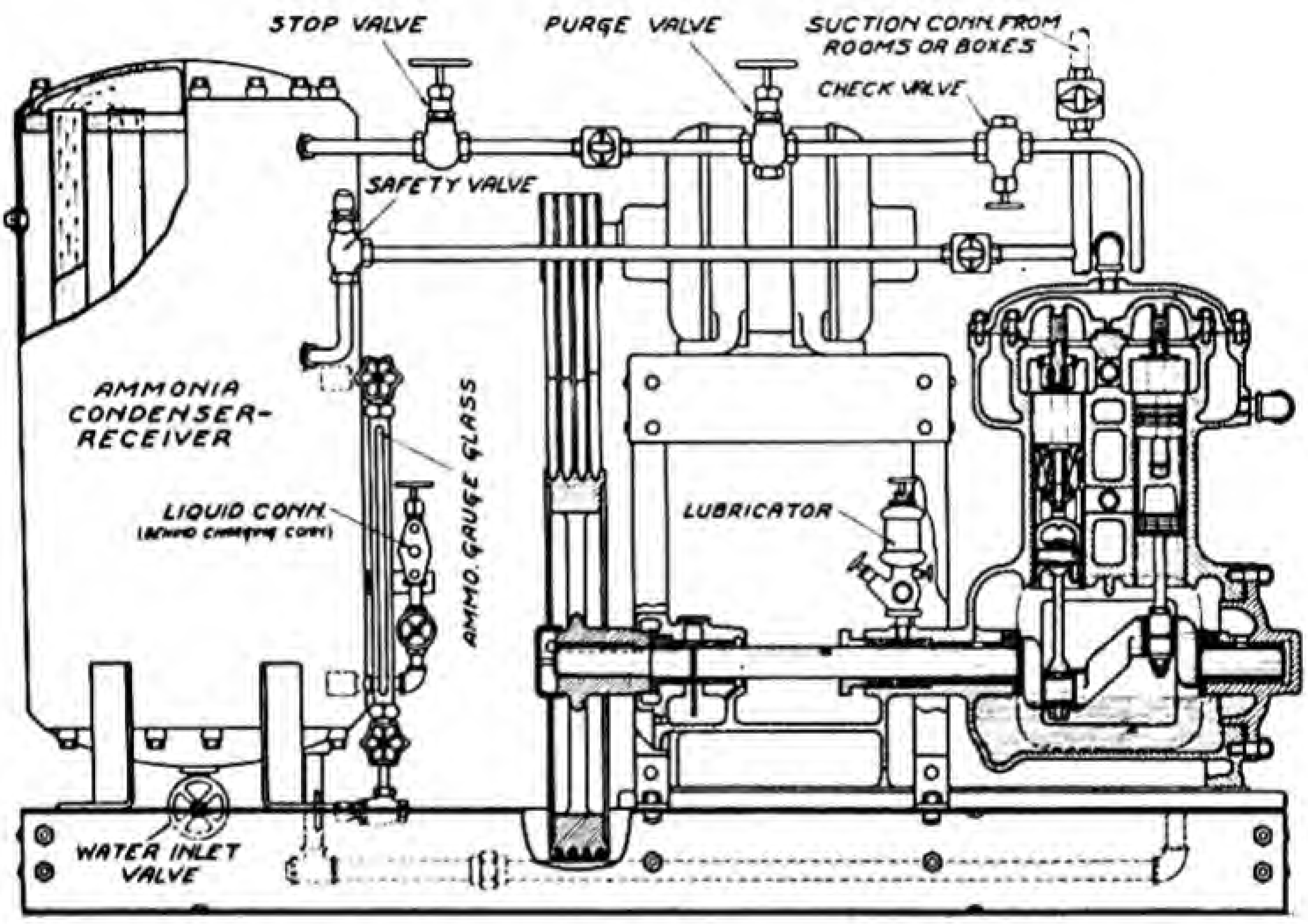


Fig. 69.—Frick Unit Type Refrigerating Machine.

Oil Interceptors.—Due to the lubricating oil becoming vaporized during compression of the ammonia, and due to particles of oil being suspended in the gas as it leaves the compressor cylinder, it is necessary to intercept as much as possible of this oil.

For the condenser to perform its work effectively, all oil must be prevented from coating the inner surface of the condenser pipe. The interceptor furnishes this safeguard to a certain extent.

The inlet of the interceptor may be arranged so that the path of the ammonia gas is changed by striking against a corrugated baffle wall. This causes the gas to travel to each side of the baffle and out

of the interceptor, while the oil drains along the baffle to the bottom of the interceptor. There is no agitation of the oil in the interceptor.

The oil may be drained off through a small valve, either globe or angle type, attached to the lowest part of the interceptor.

Fig. 70 shows the construction of welded steel oil interceptors.

The interceptor may be placed anywhere in the main discharge line between the compressor and the ammonia condenser. However, preference should be given to locating the oil interceptor as close to the condenser as possible. The interceptor is constructed entirely of wrought iron, the heads, inlet and outlet, oil drain and purge connections being welded to the body. In some interceptors oil separation is effected by causing the gas to be deflected downward in the center of the drum, thus causing the oil to settle out into the bottom of the interceptor.

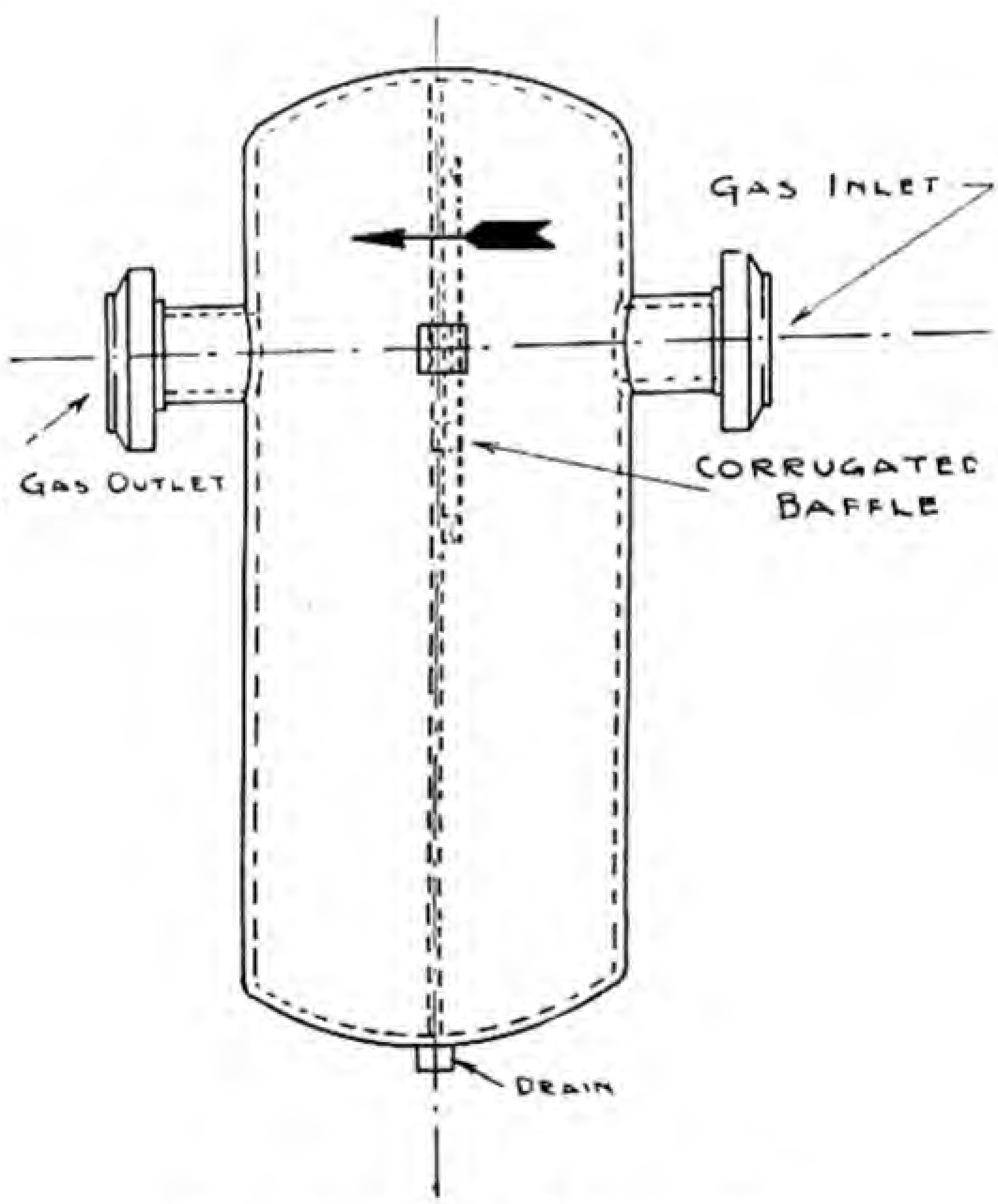


Fig. 70.—Steel Oil Interceptor.

Types of Condensers.—Ammonia condensers may be divided into four general classes, depending upon the arrangement of the cooling surface, and the manner of the flow of cooling water and the ammonia. Many new and special designs of condensers have been brought out in the last few years.

In the atmospheric type of condenser the water is led in an open flow over a vertical coil of pipe. The water is exposed to the atmosphere as it flows over the coils; thus, the name atmospheric may be well applied. The double-pipe condenser, as the name implies, is made up by inserting one pipe within another. The water flows through the internal pipe, and the ammonia through the annular space between the pipes. Surface condensers have been developed by trying to design an ammonia condenser that would be similar to the steam condenser. The shell-and-tube condenser is an example of this. The ammonia is condensed in the shell by water flowing through the tubes that extend from head to head. Another type of condenser may also be made up with a shell for holding the condensing ammonia and a series of spiral coils for the cooling water. The submerged type, as the name implies, is constructed by simply submerging an ammonia coil in water. This type is practically obsolete at present and is used only on very small plants.

The following classification indicates some of the various types of condensers which are used at present:

Class I—Atmospheric Condensers
Type A—Standard atmospheric.
Type B—Bleeder.
Type C—Flooded.
Class II—Double Pipe Condensers
Type A—Standard double-pipe.
Class III—Surface Condensers
Type A—Shell-and-tube, horizontal and vertical.
Type B—Shell-and-coil.
Class IV—Evaporative
Type A—Outdoor
Type B—Inside

The above list is probably incomplete, but contains those condensers which may be considered standard, together with several which may be considered special. The standard atmospheric, the shell-and-tube, the evaporative, and the standard double-pipe condensers are the ones that are used principally at present.

Atmospheric Condensers.—The standard atmospheric ammonia condenser is made up in the form of a flat vertical coil, and usually is 16 to 24 pipes high, and is about 20 ft. long, as illustrated in Fig. 71. Two-inch pipes are generally used and these are connected on the ends by either split or flanged return bends. In many newer installations the pipe and pipe bends are welded together. These coils are supported by iron stands and are placed on about 20-in. centers to allow sufficient passageway between them for cleaning and inspection purposes. These coils are connected by means of individual valves to suitable gas, liquid and pump-out headers.

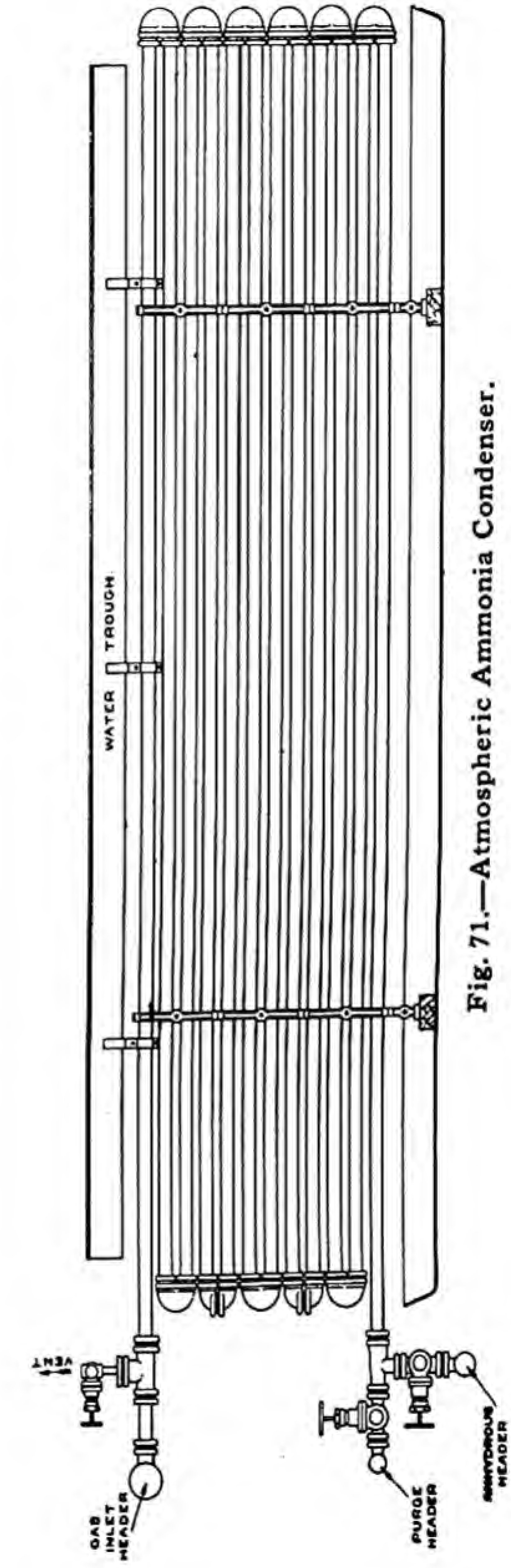


Fig. 71.—Atmospheric Ammonia Condenser.

Drip strips are sometimes placed between the pipes to guide the cooling water from one pipe to another without splashing. These strips are made of 1-in. x ⅛-in. metal and are held in position by clamps. They are not essential to the functioning of the condenser, but prevent a certain amount of the water from being wasted by splashing off the coils. They should be used where large quantities of water are used over each coil.

The cooling water is distributed over the coil surface by means of a galvanized "V" trough, or by means of a slotted pipe over each coil. The trough supports are adjustable for the purpose of elevating the trough so as to obtain an even distribution of water over the coil surface.

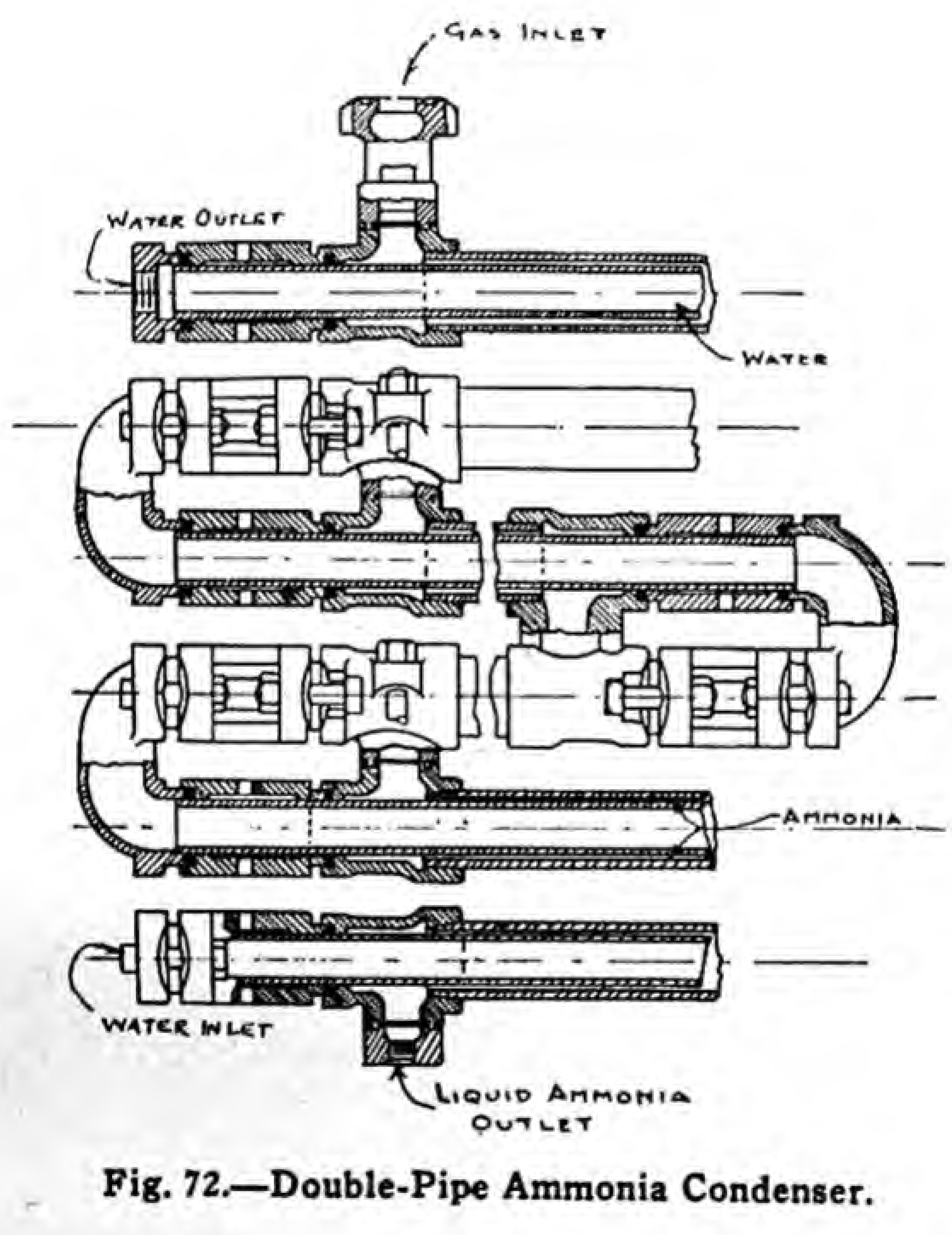


Fig. 72.—Double-Pipe Ammonia Condenser.

The hot ammonia gas from the compressor enters the coils through the gas inlet header at the top of the condenser, and flows through each pipe and downward until it reaches the bottom pipe. The water flows downward over the outside of the coil. The ammonia condenses

as it flows along the pipes until at the bottom the liquefaction has been completed. The water and ammonia flow approximately in the same direction, that is, there is no counter-current effect, but, due to the liberal allowance of coil surface and condenser water, this type of condenser really performs its functions in a satisfactory manner. Of course, in the dryer localities, the evaporation of some of the water in passing over the exposed coils aids the cooling quite appreciably. It is evident that the liquid leaving the condenser cannot be cooled to the lowest possible temperature, since it is cooled by the warm water leaving the condenser. As mentioned before there are several modifications of the atmospheric condenser but few of them continue in use at present.

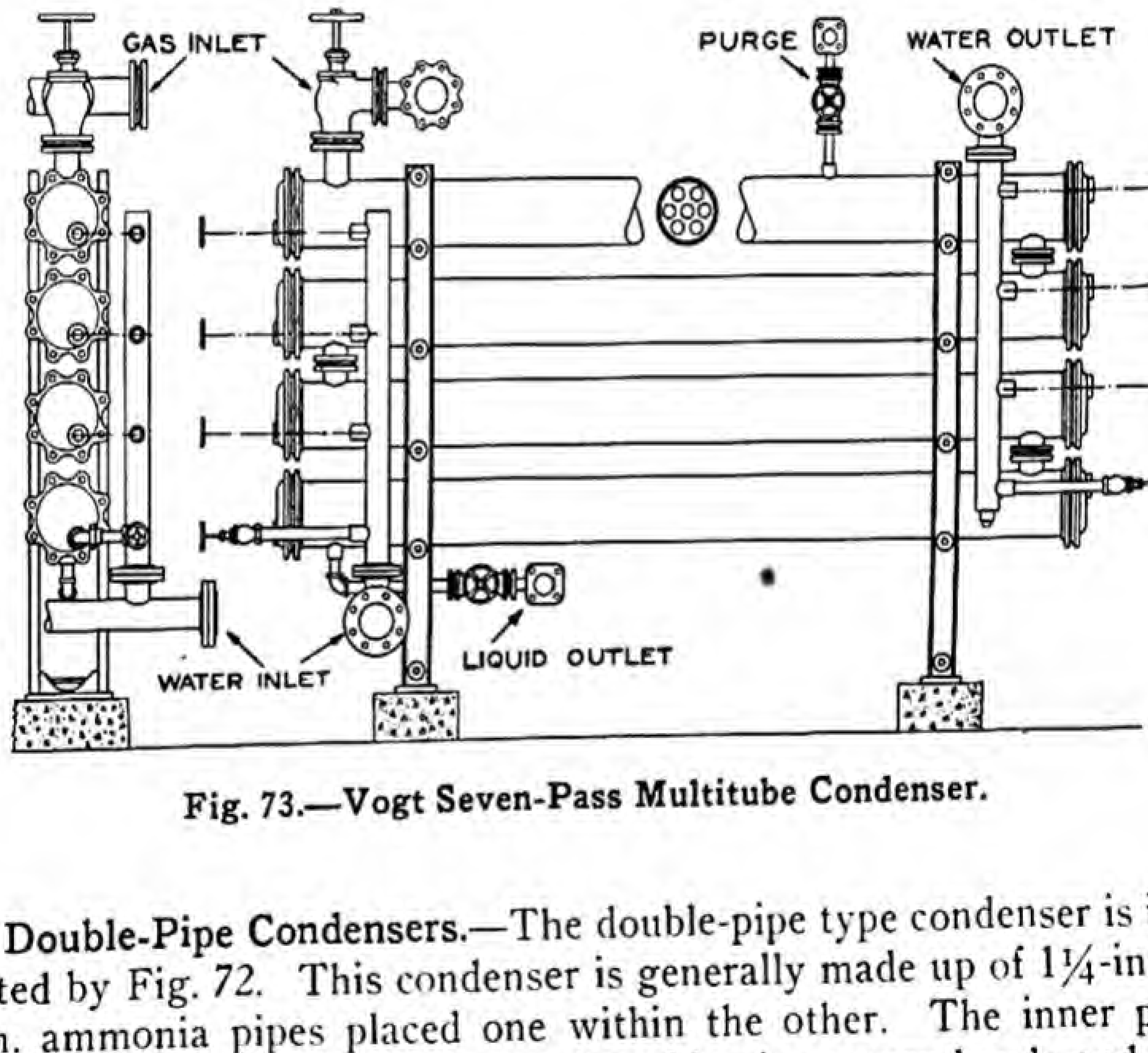


Fig. 73.—Vogt Seven-Pass Multitube Condenser.

Double-Pipe Condensers.—The double-pipe type condenser is illustrated by Fig. 72. This condenser is generally made up of 1¼-in. and 2-in. ammonia pipes placed one within the other. The inner pipes, connected at the ends by the usual double-pipe return bends and solid return bends, enclose the water circulation system. The space between the two pipes forms passages for the ammonia.

The water and ammonia flow in opposite directions, thus the condenser operates on the counter-current principle. The water enters at the bottom and passes upward through the one and one-quarter-inch pipes. The ammonia enters at the top and passes downward through the annular spaces between the pipe, leaving as liquid at the bottom. Thus, the hottest gas meets the warmest water, and during its

descent and liquefaction it becomes cooler and meets the coldest water. The ammonia liquid is cooled very near the temperature of the water at inlet. The heat transfer between the water and ammonia is good, which results in efficient operation. Also, the condenser is economical in the use of water.

Shell-and-Tube Condensers.—A horizontal shell-and-tube condenser is made up with a horizontal steel shell with heavy steel heads and a large number of extra heavy charcoal tubes that extend from head to head. The water circulation is enclosed by means of baffled heads bolted to the shell at each end. These heads cause the water to traverse the length of the condenser six to twelve times, passing, of course, through the tubes. The ammonia is retained by the shell and after liquefaction it is drained off at the bottom of the shell. Fig. 73 illustrates the construction of a 7-pass 7-tube condenser of this type.

The shell-and-tube condenser may also be arranged in a vertical position, in which case the water passes downward through the tubes and is then discharged to the sewer. In this arrangement cores are sometimes inserted into the tubes to give the water a higher velocity.

Fig. 74 shows three vertical shell-and-tube condensers. These condensers are of the single-pass water type, the water being introduced into distributing boxes located at the tops of each. Shell-and-tube condensers are simple in design and easy to clean. The vertical type occupies small floor space and provides excellent oil separation.

A vertical shell-and-spiral-coil condenser is used sometimes. The vertical shell contains the ammonia which is introduced at the top as gas, and drained off as liquid at the bottom. The cooling is accomplished by a series of spiral coils of pipe that are inserted into the shell. These coils pass through stuffing boxes in the heads and are attached to inlet and outlet water headers. This type of condenser, of course, has the disadvantage of being difficult to clean, in the case that the water would deposit scale or other matter on the coils. Principally they are used in small sizes with the low pressure refrigerants and a copper coil.

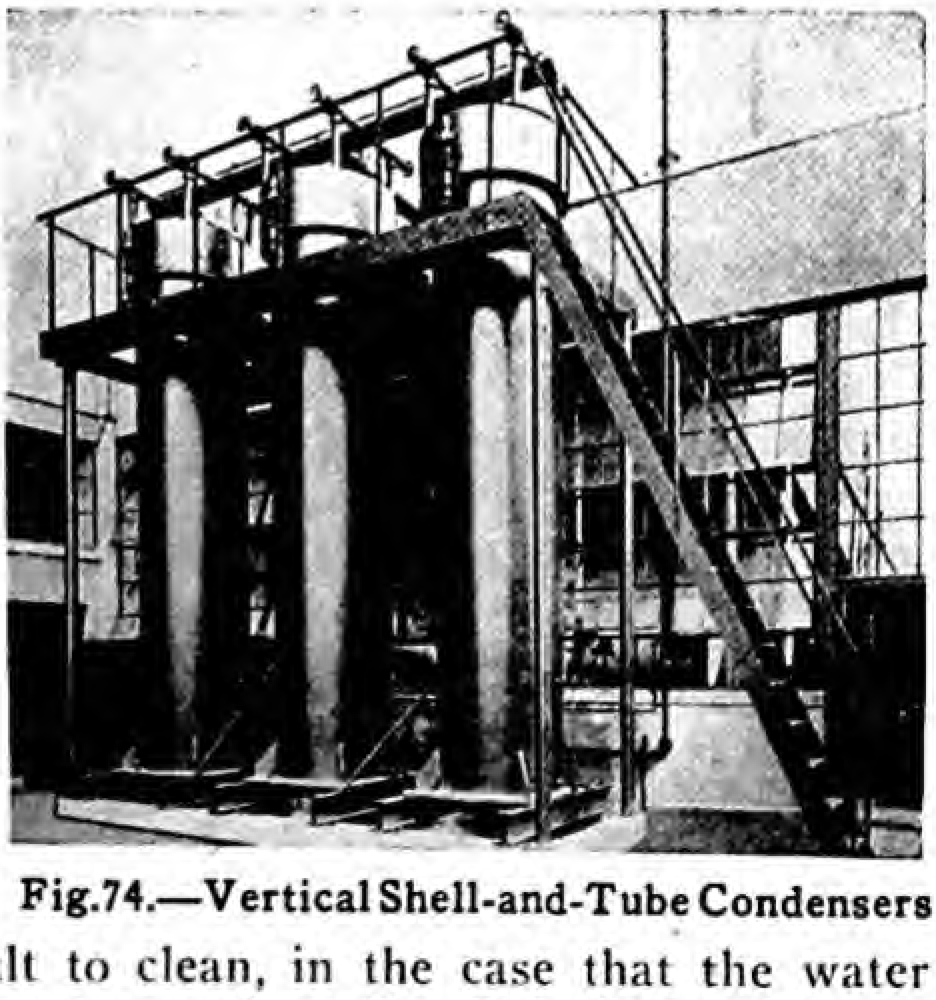


Fig.74.—Vertical Shell-and-Tube Condensers

From the foregoing discussions it will be appreciated that there are numerous ways of arranging the condenser surfaces and that the arrangement will affect the efficiency of the service of the condenser as a cooling, liquefying and aftercooling apparatus.

Evaporative Condensers.—In recent years the evaporative condenser which is a forced draft cooling tower combined with an atmospheric condenser in the same housing has become quite popular. It is very compact, moderate in cost and may be installed outside or indoors. It is described more completely in Chapter VII under cooling towers.

Liquid Receivers.—The purpose of the receiver is to provide a storage space for a large part of the refrigerant in the system. It also serves as a liquid seal. It is always desirable to have a sufficiently large receiver when a double-pipe condenser is used, since the storage space in this type of condenser is comparatively small.

Liquid receivers are made of heavy wrought iron with welded heads, and include all the regulating valves, liquid indicating gauges, etc., required for the control of the liquid. Suitable liquid inlet and outlet stop valves are provided, and a charging valve is connected with the outlet pipe to permit by-passing the receiver when charging liquid ammonia into the system. The gauge glass is protected by a suitable guard, and is equipped with controlling valves at each end. Liquid receivers are made in both the horizontal and vertical types.

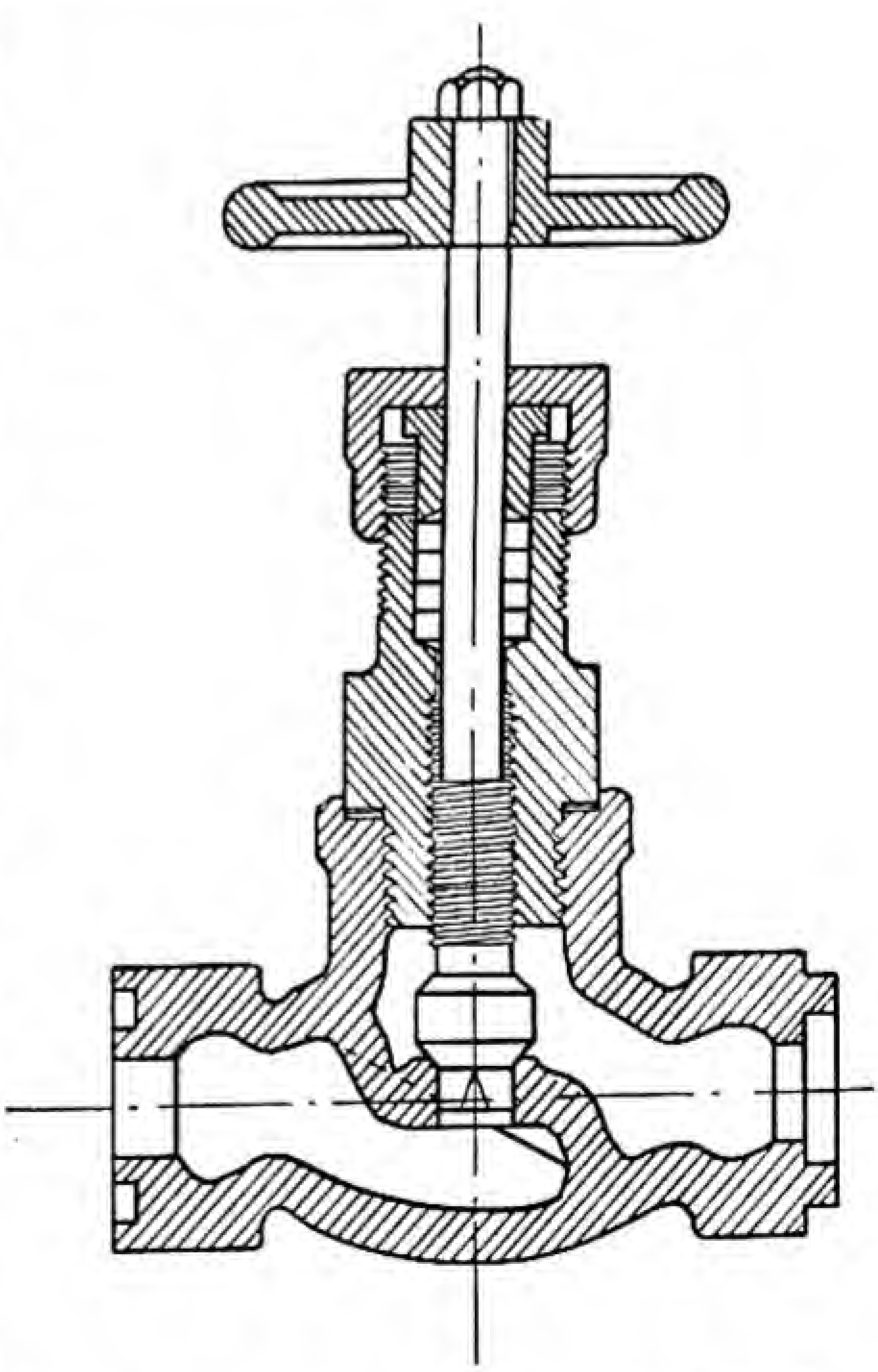


Fig. 75.—Flanged Expansion Valve.

Piping.—The successful and smooth operation of a refrigerating plant depends upon the design and layout

of the ammonia piping system. This is especially true from an operating viewpoint. Many operating difficulties may be traced to faulty piping systems.

The object of the piping system is to conduct the fluid from one apparatus to another. It is evident that this must be done with the least cost, at the same time giving proper consideration to safety requirements. Since ammonia has a corrosive and suffocating action on human life, it is necessary to make sure that the piping, valves and fittings will not fail under ordinary conditions of operation. The failure of piping, valves and fittings is not generally due to excessive pressures, but rather to the presence of liquid, vibration, temperature changes, faulty methods of support, etc.

The pressure in ammonia, gas, and liquid piping systems is fairly high, so that valves and fittings must be designed for a working pressure of 250 lbs. per sq. in. The lines and mains should be level, well aligned and free from liquid collecting traps and pockets.

Vibration in piping systems is a source of danger to all parts of the system. Vibration may be due to the intermittent flow of gas, vibration of the compressors, engines, etc. Therefore, especial attention must be given to the proper anchoring of the pipes. Similarly, the lines will contract and expand with the changes of temperature. Compensation must be made for these changes of length.

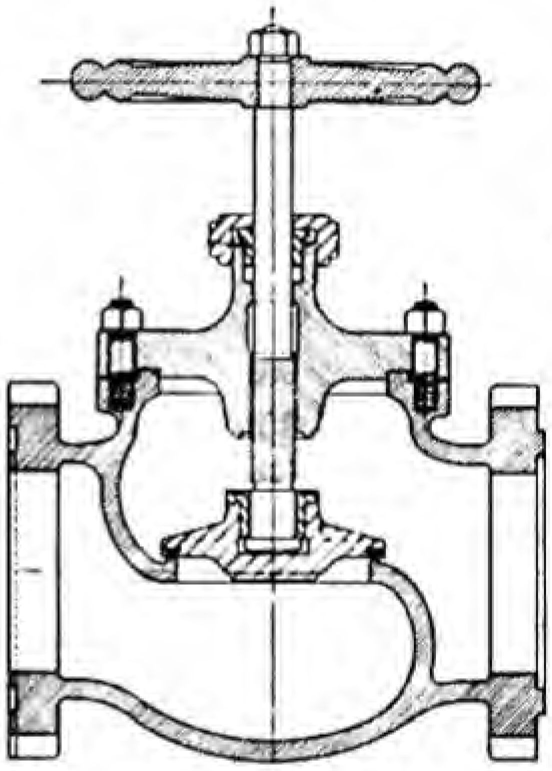


Fig.76.—Vogt Globe Valve.

Thus, the piping system and all of its components must be well designed, structurally strong, and well supported and anchored. Furthermore, it should be accessible.

Wrought iron and steel pipe may be used for ammonia connections and piping. Wrought iron pipe is more durable and also more expensive. At present steel pipe is used mostly. Liquid lines are generally made of extra heavy pipe. Discharge lines are generally made of extra heavy pipe until the larger sizes are reached. Suction lines are usually made of standard pipe, until the smaller sizes are reached. The relative cost of making pipe bends and the use of fittings have probably a more direct bearing on the weight of pipe to be used than the intensity of pressures and the size of the mains.

Pipe fittings and valves for ammonia piping systems are made of malleable iron, ferrosteel, semi-steel, drop-forged steel, etc. They are designed for 250 lbs. per sq. in. working pressures and are termed extra

heavy. These fittings may be divided into two classes, screwed and flanged. Screwed fittings are used generally on the smaller sizes only. No hard-and-fast rule may be laid down to govern the limits within which each type of fitting may be used. The tendency has been to use flanged fittings on the larger sizes. As a general rule, flanged fittings should be used on lines 2½ inches or larger.

Fig. 75 shows the construction of a small flanged expansion valve, while Fig. 76 indicates the construction of a large flanged globe valve.

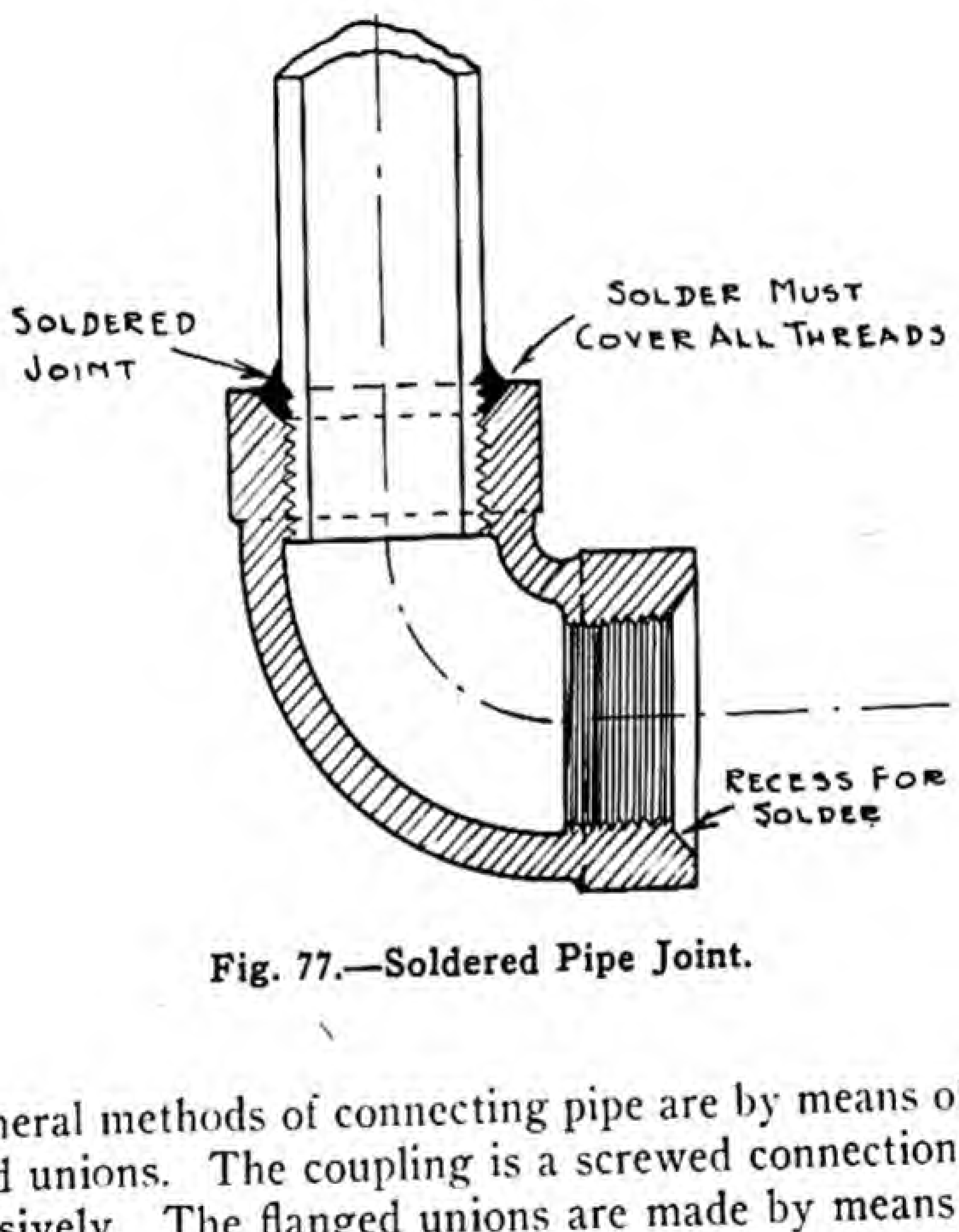


Fig. 77.—Soldered Pipe Joint.

The general methods of connecting pipe are by means of couplings and flanged unions. The coupling is a screwed connection and is not used extensively. The flanged unions are made by means of flanges, bolts and gaskets. The actual connecting of the pipe to the coupling or flanged union is accomplished by means of soldered and litharge joints. A litharge joint may be made by using a cement composed of litharge and glycerin. A soldered joint is made by covering all threads with solder and by shrinking the flange onto the pipe. The soldered joint is used more extensively at present. Litharge joints are more easily changed. Fig. 77 indicates how a screwed fitting and a pipe are connected by means of a soldered joint.

In making flanged joints, either lead, rubber, or asbestos gaskets may be used. Lead gaskets are used extensively at present.

Compressor and Capacity Data.—In the following Tables 42 through 54 data is given on refrigerant quantities, compressor displacements and capacities.

TABLE 42.—MAXIMUM SPEEDS AND DISPLACEMENTS OF TWIN V.S.A. AMMONIA COMPRESSORS.

Diam. and stroke of cylinder, ins.	Cu. in. revolution	Max r.p.m.	Piston speed ft. per min.	Displ. cu. ins. per min.
2 x 2	12.56	600	200	7,536
2½ x 2½	24.54	557	232	13,660
3 x 3	42.41	514	257	21,800
3½ x 3½	67.38	482	281	32,450
4 x 4	100.4	450	300	45,180
4½ x 4½	143.1	425	319	60,830
5 x 5	196.3	400	333	78,520
5½ x 5½	261.3	380	348	99,300
6 x 6	339.3	450	450	152,800
6½ x 6½	431.4	425	460	183,200
7 x 7	538.8	400	466	215,520
7½ x 7½	662.7	380	475	251,800
8 x 8	804.3	360	480	289,700
8½ x 8½	964.7	327	463	315,500
9 x 9	1145.1	327	490	375,000
9½ x 9½	1346.8	300	475	404,000
10 x 10	1570.8	300	500	471,240
10½ x 10½	1818.4	277	485	503,500
12 x 12	2714.4	257	514	697,000

NOTE—Sizes 2"x2" to 5½"x5½", splash lubrication, generally.
NOTE—Sizes 6"x6" to 12"x12", forcefeed lubrication, generally.

Horizontal Ammonia Compressors.—The maximum allowable revolutions per minute for slow speed ammonia compressors may be determined by Gardner's rule as follows:

$$\text{Max. r.p.m.} = \frac{376}{\sqrt{s}}$$

Where s = stroke of piston, in inches.

Similarly, the maximum allowable revolutions per min. for high-speed ammonia compressors may be determined as follows:

$$\text{r.p.m.h.s.} = \frac{850}{\sqrt{s}}$$

TABLE 43.—REFRIGERATING CAPACITIES AND BRAKE HORSEPOWER OF AMMONIA COMPRESSORS AT MAXIMUM SPEEDS SHOWN IN TABLE 42.

Comp.	0#S.P.		10#S.P.		20#S.P.		30#S.P.		40#S.P.	
Size	Tons Refr.	B.H.P.	Tons Refr.	B.H.P.	Tons Refr.	B.H.P.	Tons Refr.	B.H.P.	Tons Refr.	B.H.P.
2 x2	0.36	1.32	0.62	1.64	0.96	1.82	1.20	1.94	1.46	1.96
2½x2½	0.65	2.40	1.13	2.98	1.74	3.31	2.18	3.52	2.65	3.55
3 x3	1.16	3.66	2.03	4.65	2.93	5.22	3.86	5.60	4.74	5.67
3½x3½	1.72	5.45	3.02	6.93	4.36	7.78	5.75	8.34	7.06	8.44
4 x4	2.41	7.18	4.22	9.05	6.02	10.10	8.00	10.75	9.80	10.94
4½x4½	3.25	9.67	5.69	12.20	8.11	13.61	10.77	14.50	13.20	14.75
5 x5	4.21	11.88	7.33	15.10	10.52	16.87	13.88	17.98	16.97	18.31
5½x5½	5.32	15.03	9.27	19.10	13.30	21.32	17.55	22.72	21.47	23.15
6 x6	8.10	22.20	14.20	28.60	20.65	32.20	27.00	34.00	33.30	34.70
6½x6½	9.71	26.50	17.02	34.30	24.75	38.60	32.40	40.75	39.90	41.60
7 x7	11.50	31.20	20.10	39.60	29.00	44.40	38.10	47.00	46.80	47.80
7½x7½	13.43	36.50	23.50	46.30	33.90	51.90	44.50	54.90	54.70	55.85
8 x8	15.20	40.70	26.80	52.00	39.70	58.50	51.00	62.70	63.00	64.00
8½x8½	16.55	44.30	29.20	56.60	43.20	63.70	55.60	68.30	68.10	69.70
9 x9	19.70	52.50	34.90	66.80	50.30	74.60	66.50	80.60	82.00	81.30
9½x9½	21.20	56.60	37.60	72.00	54.20	80.20	71.70	86.80	88.30	87.60
10x10	25.10	66.50	43.90	84.00	63.50	94.00	83.60	102.00	103.10	102.80
10½x10½	26.80	71.00	46.60	89.60	67.80	100.20	89.20	108.80	110.00	109.80
12x12	37.20	98.40	64.60	124.10	94.00	138.80	123.50	150.80	152.50	152.30

For a condenser pressure of 185 lbs. gauge; dry compression with no liquid subcooling.

TABLE 44.—MAXIMUM SPEEDS AND DISPLACEMENTS OF SINGLE CYLINDER H.D.A. AMMONIA COMPRESSORS.

Diam. and stroke of cylinder, ins.	Cu. in. revolution	Max. r.p.m.	Piston speed ft. per min.	Displ. cu. ins. per min.
9x 9	1145.1	327	490	375,000
10x10	1570.8	300	500	471,240
11x11	2090.7	277	508	579,000
12x12	2714.4	257	514	697,000
13x13	3451.0	240	520	829,000
15x15	5301.3	225	562	1,193,000
17x17	8717.3	200	567	1,743,000
19x19	10774.1	180	570	1,951,000
21x21	14547.1	164	574	2,383,000
24x24	21714.7	150	600	3,260,000

TABLE 45.—REFRIGERATING EFFECT OF FREON-12 IN BTU. PER LB. DRY COMPRESSION WITH NO LIQUID SUBCOOLING.

Gauge Pressure Lbs. per Sq. In.	Saturated Evaporating Temp. °F.	Condensing Gauge Pressure											
		57.71	63.73	70.12	76.90	84.06	91.6	93.2	99.6	108.0	116.9	126.2	136.0
		Saturated Condensing Temperature °F.											
		60	65	70	75	80	85	86	90	95	100	105	110
10.92*	−40	51.93	50.78	49.60	48.42	47.22	46.02	45.78	44.80	43.57	42.34	41.10	39.85
8.34*	−35	52.43	51.28	50.10	48.92	47.72	46.52	46.28	45.30	44.07	42.84	41.60	40.35
5.45*	−30	53.13	51.98	50.80	49.62	48.42	47.22	46.98	46.00	44.77	43.57	42.30	41.05
2.28*	−25	53.73	52.58	51.40	50.22	49.02	47.82	47.58	46.60	45.37	44.14	42.90	41.65
0.58	−20	54.30	53.15	51.97	50.79	49.59	48.39	48.15	47.17	45.94	44.71	43.47	42.20
2.46	−15	54.88	53.73	52.55	51.37	50.17	48.97	48.73	47.75	46.52	45.29	44.05	42.80
4.50	−10	55.48	54.33	53.15	51.97	50.77	49.57	49.33	48.35	47.12	45.89	44.65	43.40
6.74	− 5	56.06	54.91	53.73	52.55	51.35	50.15	49.91	48.93	47.70	46.47	45.23	43.98
9.17	0	56.64	55.49	54.31	53.13	51.98	50.73	50.49	49.51	48.28	47.05	45.81	44.56
11.81	5	57.22	56.07	54.89	53.71	52.51	51.31	51.07	50.09	48.86	47.63	46.39	45.14
14.65	10	57.79	56.64	55.46	54.28	52.98	51.88	51.64	50.66	49.43	48.20	46.96	45.71
17.74	15	58.36	57.21	56.03	54.85	53.65	52.45	52.21	51.23	50.00	48.77	47.53	46.28
21.05	20	58.92	57.77	56.59	55.41	54.21	53.01	52.77	51.79	50.56	49.33	48.09	46.84
24.62	25	59.49	58.34	57.16	55.98	54.78	53.58	53.34	52.36	51.13	49.90	48.66	47.41
28.46	30	60.04	58.89	57.71	56.53	55.33	54.13	53.89	52.91	51.68	50.45	49.21	47.96
32.57	35	60.59	59.44	58.26	57.08	55.88	54.68	54.44	53.46	52.23	51.00	49.76	48.51
36.98	40	61.14	59.99	58.81	57.63	56.43	55.23	54.99	54.01	52.78	51.55	50.31	49.16

*Inches of mercury below one atmosphere.

TABLE 46.—POUNDS OF FREON-12 PER MINUTE PER TON OF REFRIGERATION DRY COMPRESSION WITH NO LIQUID SUBCOOLING.

Gauge Pressure Lbs. per Sq. In.	Saturated Evaporating Temp. °F.	Condensing Gauge Pressure											
		57.71	63.73	70.12	76.90	84.06	91.6	93.2	99.6	108.0	116.9	126.2	136.0
		Saturated Condensing Temperature °F.											
		60	65	70	75	80	85	86	90	95	100	105	110
10.92*	−40	3.86	3.94	4.04	4.13	4.24	4.35	4.37	4.47	4.59	4.72	4.87	5.02
8.34*	−35	3.81	3.90	3.99	4.09	4.19	4.30	4.32	4.42	4.54	4.67	4.81	4.96
5.45*	−30	3.77	3.85	3.93	4.03	4.13	4.24	4.26	4.35	4.47	4.59	4.73	4.88
2.28*	−25	3.73	3.81	3.89	3.98	4.08	4.18	4.20	4.29	4.41	4.53	4.66	4.81
0.58	−20	3.69	3.77	3.85	3.94	4.04	4.13	4.16	4.24	4.36	4.47	4.60	4.74
2.46	−15	3.64	3.72	3.81	3.90	3.99	4.09	4.11	4.19	4.30	4.42	4.54	4.68
4.50	−10	3.60	3.69	3.77	3.85	3.95	4.04	4.06	4.14	4.24	4.36	4.48	4.61
6.74	− 5	3.57	3.65	3.73	3.81	3.90	3.99	4.01	4.08	4.20	4.31	4.42	4.55
9.17	0	3.53	3.61	3.69	3.77	3.85	3.95	3.97	4.04	4.14	4.25	4.36	4.49
11.81	5	3.49	3.57	3.64	3.73	3.81	3.90	3.92	3.99	4.09	4.20	4.31	4.43
14.65	10	3.46	3.53	3.60	3.68	3.78	3.85	3.87	3.95	4.05	4.15	4.26	4.38
17.74	15	3.43	3.50	3.57	3.65	3.73	3.81	3.83	3.90	4.00	4.10	4.21	4.32
21.05	20	3.40	3.47	3.53	3.61	3.68	3.77	3.79	3.87	3.96	4.05	4.16	4.27
24.62	25	3.37	3.43	3.50	3.58	3.65	3.74	3.76	3.82	3.91	4.01	4.11	4.22
28.46	30	3.33	3.40	3.47	3.54	3.62	3.70	3.71	3.78	3.87	3.96	4.07	4.18
32.57	35	3.30	3.36	3.43	3.51	3.58	3.66	3.67	3.74	3.83	3.92	4.02	4.13
36.98	40	3.27	3.33	3.40	3.47	3.54	3.62	3.64	3.70	3.79	3.88	3.98	4.07

* Inches of mercury below one atmosphere.

TABLE 47.—THEORETICAL DISPLACEMENTS OF F-12 COMPRESSORS CU. INS. PER MIN. PER TON REFRIGERATION DRY COMPRESSION WITH NO LIQUID SUBCOOLING.

Gauge Pressure Lbs. per Sq. In.	Saturated Evaporating Temp. °F.	Condensing Gauge Pressure											
		57.71	63.73	70.12	76.90	84.06	91.6	93.2	99.6	108.0	116.9	126.2	136.0
		Saturated Condensing Temperature °F.											
		60	65	70	75	80	85	86	90	95	100	105	110
10.92*	−40	26100	26600	27300	27900	28650	29400	29500	30200	31000	31900	32900	33900
8.34*	−35	22850	23400	23900	24500	25100	25800	25900	26500	27200	28000	28850	29750
5.45*	−30	20100	20500	20950	21500	22000	22600	22700	23200	23850	24500	25200	26000
2.28*	−25	17700	18150	18550	18950	19450	19900	20000	20450	21000	21600	22200	22900
0.58	−20	15750	16100	16450	16800	17250	17650	17750	18100	18600	19100	19650	20250
2.46	−15	14000	14300	14650	15000	15350	15750	15800	16100	16550	17000	17450	18000
4.50	−10	12450	12750	13050	13330	13650	13980	14040	14330	14660	15070	15500	15950
6.74	− 5	11150	11400	11650	11900	12180	12470	12530	12750	13130	13470	13880	14220
9.17	0	10000	10220	10430	10660	10880	11170	11230	11430	11720	12030	12330	12700
11.81	5	8880	9160	9340	9570	9780	10000	10050	10230	10480	10770	11050	11350
14.65	10	8080	8240	8400	8590	8820	8980	9040	9220	9460	9690	9940	10230
17.74	15	7290	7440	7590	7760	7930	8100	8140	8290	8500	8710	8950	9180
21.05	20	6580	6720	6840	7000	7130	7300	7340	7500	7670	7840	8060	8270
24.62	25	5970	6080	6200	6340	6470	6630	6660	6770	6930	7110	7280	7480
28.46	30	5410	5520	5630	5740	5870	6000	6020	6140	6280	6420	6600	6780
32.57	35	4635	4720	4820	4930	5030	5140	5160	5260	5380	5500	5650	5800
36.98	40	4475	4555	4650	4750	4845	4950	4980	5060	5180	5310	5440	5570

* Inches of mercury below one atmosphere.

TABLE 48.—REFRIGERATING EFFECT OF SULPHUR DIOXIDE BTU. PER LB. DRY COMPRESSION WITH NO LIQUID SUBCOOLING.

Gauge Pressure Lbs. per Sq. In.	Saturated Evaporating Temp. °F.	Condensing Gauge Pressure											
		26.23	30.43	34.92	39.77	44.98	50.58	51.75	56.55	62.90	69.82	77.15	85.06
		Saturated Condensing Temperature °F.											
		60	65	70	75	80	85	86	90	95	100	105	110
23.54*	−40	145.5	143.8	142.0	140.3	138.6	136.8	136.5	135.1	133.4	131.7	129.7	128.3
22.41*	−35	146.2	144.4	142.7	141.0	139.2	137.5	137.1	135.8	134.1	132.5	130.4	129.0
21.10*	−30	146.8	145.1	143.3	141.6	139.8	138.1	137.8	136.4	134.7	133.0	131.0	129.6
19.63*	−25	147.4	145.7	143.9	142.2	140.4	138.7	138.4	137.0	135.3	133.6	131.6	130.2
17.93*	−20	148.0	146.2	144.5	142.7	141.0	139.3	138.9	137.6	135.9	134.2	132.2	130.8
16.05*	−15	148.5	146.8	145.0	143.3	141.6	139.8	139.5	138.1	136.4	134.7	132.7	131.4
13.91*	−10	149.0	147.3	145.6	143.8	142.1	140.3	140.0	138.6	136.9	135.2	133.3	131.9
11.52*	− 5	149.5	147.8	146.0	144.3	142.6	140.8	140.5	139.1	137.4	135.7	133.7	132.4
8.85*	0	150.0	148.2	146.5	144.7	143.0	141.3	140.9	139.6	137.9	136.2	134.2	132.8
5.87*	5	150.4	148.7	146.9	145.2	143.4	141.7	141.4	140.0	138.3	136.6	134.6	133.2
2.59*	10	150.8	149.0	147.3	145.5	143.8	142.1	141.9	140.4	138.7	137.0	135.0	133.6
0.51	15	151.1	149.4	147.6	145.9	144.2	142.4	142.1	140.7	139.0	137.3	135.3	133.9
2.48	20	151.4	149.7	147.9	146.2	144.5	142.7	142.4	141.0	139.3	137.6	135.6	134.3
4.64	25	151.7	149.9	148.2	146.5	144.6	143.0	142.8	141.3	139.6	137.9	135.9	134.5
7.00	30	151.9	150.2	148.4	146.7	145.0	143.2	142.9	141.5	139.8	138.1	136.1	134.8
9.58	35	152.1	150.4	148.6	146.9	145.2	143.5	143.1	141.7	140.0	138.2	136.3	135.0
12.40	40	152.3	150.5	148.8	147.0	145.3	143.6	143.3	141.9	140.2	138.5	136.5	135.1

* Inches of mercury below one atmosphere.

TABLE 49.—POUNDS OF SULPHUR DOXIDE PER MINUTE PER TON REFRIGERATION DRY COMPRESSION WITH NO LIQUID SUBCOOLING.

Gauge Pressure Lbs. per Sq In.	Saturated Evaporating Temp. °F.	Condensing Gauge Pressure											
		26.23	30.43	34.92	39.77	44.98	50.58	51.75	56.55	62.90	69.82	77.15	85.06
		Saturated Condensing Temperature °F.											
		60	65	70	75	80	85	86	90	95	100	105	110
23.54*	−40	1.374	1.391	1.408	1.426	1.443	1.462	1.465	1.480	1.499	1.518	1.542	1.558
22.41*	−35	1.368	1.385	1.402	1.419	1.436	1.455	1.458	1.473	1.492	1.510	1.534	1.550
21.10*	−30	1.362	1.379	1.395	1.413	1.430	1.448	1.452	1.466	1.485	1.504	1.526	1.543
19.63*	−25	1.357	1.373	1.390	1.407	1.424	1.441	1.445	1.460	1.478	1.497	1.519	1.536
17.93*	−20	1.352	1.368	1.384	1.401	1.418	1.436	1.439	1.454	1.472	1.491	1.513	1.528
16.05*	−15	1.346	1.363	1.379	1.396	1.413	1.430	1.434	1.448	1.466	1.485	1.507	1.522
13.91*	−10	1.342	1.358	1.374	1.390	1.408	1.425	1.428	1.442	1.460	1.479	1.501	1.517
11.52*	− 5	1 338	1.353	1.369	1.386	1.403	1.420	1.423	1.438	1.455	1.474	1.495	1.511
8.85*	0	1.334	1.349	1.365	1.383	1.398	1.415	1.420	1.433	1.450	1.469	1.490	1.505
5.87*	5	1.330	1.345	1.361	1.378	1.394	1.411	1.415	1.429	1.446	1.464	1.485	1.501
2.59*	10	1.327	1.342	1.356	1.374	1.390	1.408	1.410	1.425	1.442	1.460	1.482	1.497
0.51	15	1.323	1.339	1.354	1.371	1.387	1.404	1.408	1.421	1.438	1.457	1.480	1.491
2.48	20	1.321	1.336	1.352	1.368	1.384	1.401	1.404	1.418	1.435	1.453	1.474	1.489
4.64	25	1.319	1.334	1.349	1.365	1.382	1.398	1.400	1.415	1.433	1.450	1.471	1.487
7.00	30	1.316	1.332	1.347	1.363	1.380	1.396	1.399	1.413	1.430	1.448	1.469	1.484
9.58	35	1.315	1.330	1.345	1.361	1.378	1.395	1.398	1.411	1.428	1.446	1.467	1.482
12.40	40	1.313	1.328	1.344	1.360	1.376	1.393	1.396	1.410	1.427	1.444	1.465	1.480

* Inches of mercury below one atmosphere.

TABLE 50.—THEORETICAL DISPLACEMENT OF SULPHUR DIOXIDE COMPRESSORS CU. INS. PER MIN. PER TON REFRIGERATION DRY COMPRESSION WITH NO LIQUID SUBCOOLING.

Gauge Pressure Lbs. per Sq. In.	Saturated Evaporating Temp. °F.	Condensing Gauge Pressure											
		26.23	30.43	34.92	39.77	44.98	50.58	51.75	56.55	62.90	69.82	77.15	85.06
		Saturated Condensing Temperature °F.											
		60	65	70	75	80	85	86	90	95	100	105	110
23.54*	−40	53210	53890	54560	55230	55920	56630	56770	57350	58080	58830	59730	60370
22.41*	−35	45470	46010	46570	47150	47730	48340	48460	48950	49570	50170	50970	51510
21.10*	−30	38990	39450	39930	40420	40920	41430	41540	41960	42490	43030	43680	44140
19.63*	−25	33550	33950	34360	34780	35210	35630	35740	36100	36550	37020	37570	37970
17.93*	−20	29010	29350	29710	30070	30440	30820	30890	31200	31590	31990	32470	32790
16.05*	−15	25140	25450	25750	26070	26390	26710	26780	27050	27390	27730	28140	28440
13.91*	−10	21890	22150	22410	22670	22960	23240	23300	23530	23830	24130	24480	24740
11.52*	− 5	19140	19360	19590	19830	20070	20320	20370	20570	20820	21080	21400	21620
8.85*	0	16770	16970	17170	17390	17590	17810	17860	18030	18250	18480	18750	18930
5.87*	5	14750	14930	15100	15290	15470	15660	15700	15850	15920	16250	16470	16660
2.59*	10	13030	13180	13330	13490	13650	13820	13940	13990	14160	14340	14550	14700
0.51	15	11530	11670	11800	11940	12090	12230	12260	12380	12530	12690	12870	13010
2.48	20	10240	10360	10480	10600	10730	10860	10890	11000	11130	11270	11430	11550
4.64	25	9100	9205	9310	9424	9543	9652	9662	9769	9890	10010	10160	10260
7.00	30	8146	8241	8337	8436	8537	8640	8661	8745	8851	8960	9090	9180
9.58	35	7300	7382	7465	7557	7647	7740	7757	7833	7928	8025	8142	8225
12.40	40	6550	6628	6706	6785	6866	6948	6955	7033	7118	7206	7310	7385

* Inches of mercury below one atmosphere.

TABLE 51.—REFRIGERATING CHARACTERISTICS OF METHYL CHLORIDE. REFRIGERATING EFFECT IN BTU. PER LB.; LBS. EVAPORATED PER MIN. PER TON OF REFRIGERATION; THEORETICAL COMPRESSOR DISPLACEMENT IN CU. INS. PER TON REFRIGERATION.

Gauge Pressure Lbs. per Sq. In.	Saturated Evaporating Temp. °F.		Condensing Gauge Pressure						
			46.3	57.8	72.3	80.8	87.3	102.3	118.3
			Saturated Condensing Temperature °F.						
			60	70	80	86	90	100	110
6.51*	−20	Refrigerating effect	152.5	148.5	145.5	142.4	141.0	137.0	133.5
		Lbs. per min. per ton	1.311	1.347	1.375	1.404	1.418	1.460	1.498
		Displacement cu. ins.	17510	17980	18360	18760	18940	19490	20010
0.41*	−10	Refrigerating effect	155.0	151.0	147.0	144.9	143.5	139.5	136.0
		Lbs. per min. per ton	1.290	1.324	1.360	1.380	1.394	1.434	1.470
		Displacement cu. ins.	13810	14170	14560	14770	14910	15340	15730
3.8	0	Refrigerating effect	157.5	153.5	150.5	147.4	146.0	142.0	138.5
		Lbs. per min. per ton	1.270	1.303	1.329	1.357	1.370	1.408	1.444
		Displacement cu. ins.	10980	11270	11490	11730	11850	12180	12490
6.19	5	Refrigerating effect	158.8	154.8	150.8	148.7	147.3	143.3	139.8
		Lbs. per min. per ton	1.259	1.292	1.326	1.345	1.358	1.396	1.431
		Displacement cu. ins.	9856	10110	10380	10530	10630	10920	11200
8.6	10	Refrigerating effect	160.0	156.0	152.0	149.9	148.5	144.5	141.0
		Lbs. per min. per ton	1.250	1.282	1.316	1.334	1.347	1.384	1.418
		Displacement cu. ins.	8845	9071	9310	9441	9530	9794	10040
13.6	20	Refrigerating effect	163.5	158.5	155.5	152.4	151.0	147.0	143.5
		Lbs. per min. per ton	1.223	1.262	1.286	1.312	1.324	1.360	1.394
		Displacement cu. ins.	7117	7341	7483	7635	7706	7915	8110
20.3	30	Refrigerating effect	164.7	160.7	156.7	154.6	153.2	149.2	145.7
		Lbs. per min. per ton	1.214	1.247	1.276	1.294	1.305	1.340	1.373
		Displacement cu. ins.	5832	5978	6130	6214	6270	6438	6593
28.1	40	Refrigerating effect	167.2	163.2	159.2	157.1	155.7	151.7	148.2
		Lbs. per min. per ton	1.196	1.225	1.256	1.273	1.284	1.318	1.349
		Displacement cu. ins.	4794	4912	5035	5103	5148	5284	5409

* Inches of mercury below one atmosphere.

TABLE 52.—REFRIGERATING EFFECT OF CARBON DIOXIDE
DRY COMPRESSION WITH NO LIQUID SUBCOOLING.

Gauge Pressure Lbs. per Sq. In.	Saturated Evaporating Temp. °F.	Condensing Gauge Pressure						
		744	785	839	895	954	1016	1028
		Saturated Condensing Temperature °F.						
		60	65	70	75	80	85	86
131	−40	82.3	78.4	74.1	69.4	63.9	56.5	54.5
147	−35	82.5	78.6	74.3	69.6	64.1	56.7	54.7
163	−30	82.7	78.8	74.5	69.8	64.3	56.9	54.9
181	−25	82.8	78.9	74.6	69.9	64.4	57.0	55.0
200	−20	83.0	79.1	74.8	70.1	64.6	57.2	55.2
221	−15	83.2	79.3	75.0	70.3	64.8	57.4	55.4
243	−10	83.2	79.3	75.0	70.3	64.8	57.4	55.4
266	− 5	83.3	79.4	75.1	70.4	64.9	57.5	55.4
291	0	83.4	79.5	75.2	70.5	65.0	57.6	55.6
317	5	83.2	79 3	75.0	70.3	64.8	57.4	55.4
345	10	83.2	79.3	75.0	70.3	64.8	57.4	55.4
375	15	83.1	79.2	74.9	70.2	64.7	57.3	55.3
407	20	82.8	78.9	74.6	69.9	64.4	57.0	55.0
435	25	82 5	78.6	74.3	69.6	64.1	56.7	54.7
476	30	82.3	78.4	74.1	69.4	63.9	56.5	54.5
514	35	81.8	77.9	73.6	68.9	63.4	56.0	54.0
553	40	81.2	77.3	73.0	68.3	62.8	55.4	53.4

TABLE 53.—POUNDS OF CARBON DIOXIDE EVAPORATED PER MIN. PER TON OF REFRIGERATION DRY COMPRESSION WITH NO LIQUID SUBCOOLING

Gauge Pressure Lbs. per Sq. In.	Saturated Evaporating Temp. °F.	Condensing Gauge Pressure						
		744	785	839	895	954	1016	1028
		Saturated Condensing Temperature °F.						
		60	65	70	75	80	85	86
131	−40	2.430	2.550	2.700	2.880	3.130	3.540	3.670
147	−35	2.425	2.547	2.690	2.875	3.120	3.530	3.660
163	−30	2.420	2.540	2.687	2.867	3.110	3.520	3.650
181	−25	2.418	2.538	2.682	2.863	3.108	3.510	3.640
200	−20	2.410	2.530	2.676	2.855	3.100	3.500	3.625
221	−15	2.405	2.523	2.670	2.850	3.090	3.490	3.613
243	−10	2.405	2.523	2.670	2.850	3.090	3.490	3.613
266	− 5	2.400	2.520	2.665	2.842	3.085	3.480	3.610
291	0	2.398	2.516	2.660	2.838	3.078	3.475	3.600
317	5	2.405	2.523	2.670	2.850	3.090	3.490	3.613
345	10	2.405	2.523	2.670	2.850	3.090	3.490	3.613
375	15	2.410	2.527	2.673	2.851	3.092	3.492	3.617
407	20	2.418	2.538	2.682	2.863	3.108	3.510	3.640
435	25	2.425	2.547	2.690	2.875	3.120	3.530	3.660
476	30	2.430	2.550	2.700	2.880	3.130	3.540	3.670
514	35	2.447	2.570	2.720	2.903	3.158	3.574	3.707
553	40	2.465	2.588	2.742	2.930	3.186	3.610	3.748

TABLE 54.—THEORETICAL DISPLACEMENTS OF CARBON DIOXIDE COMPRESSORS CU. IN. PER MIN. PER TON OF REFRIGERATION DRY COMPRESSION WITH NO LIQUID SUBCOOLING.

Gauge Pressure Lbs. per Sq. In.	Saturated Evaporating Temp. °F.	Condensing Gauge Pressure						
		744	785	839	895	954	1016	1028
		Saturated Condensing Temperature °F.						
		60	65	70	75	80	85	86
131	−40	2565	2692	2850	3040	3303	3735	3873
147	−35	2323	2440	2577	2755	2990	3380	3506
163	−30	2104	2210	2335	2492	2704	3060	3170
181	−25	1914	2010	2122	2265	2458	2776	2880
200	−20	1737	1824	1930	2057	2235	2522	2610
221	−15	1567	1645	1740	1857	2013	2275	2355
243	−10	1443	1513	1602	1710	1855	2093	2170
266	− 5	1320	1385	1464	1563	1695	1912	1985
291	0	1203	1263	1335	1423	1543	1743	1806
317	5	1106	1162	1228	1312	1422	1605	1660
345	10	1012	1063	1124	1199	1301	1301	1520
375	15	931	976	1033	1102	1195	1348	1396
407	20	956	898	950	1014	1100	1243	1288
435	25	788	827	874	934	1013	1146	1188
476	30	723	759	803	856	931	1053	1092
514	35	667	701	741	791	861	974	1010
553	40	616	646	685	732	794	902	935

QUESTIONS ON CHAPTER VI.

1. What are some of the advantages of the horizontal double-acting ammonia compressor?

2. Name some of the advantages of the vertical single-acting ammonia compressor.

3. Describe the important characteristics of the atmospheric type ammonia condenser.

4. Name some of the advantages of the shell-and-tube type of condenser.

5. What are some of the features of the double-pipe ammonia condenser?

6. What is the refrigerating capacity of a 15-in. diam. by 15-in. stroke horizontal double-acting ammonia compressor operating at 225 r.p.m. between 15 lbs. gauge suction pressure and 200 lbs. gauge condenser pressure when the volumetric efficiency is 67 per cent?

7. What is the refrigerating capacity of an 18-in. diam. by 30-in. stroke horizontal double-acting ammonia compressor operating at 70 r.p.m. between 20 lbs. gauge suction pressure and 185 lbs. gauge condenser pressure when the volumetric efficiency is 85 per cent?

8. What is the indicated horsepower of the compressor described in Problem 7?

9. What factors determine the horsepower and the refrigerating capacity of an ammonia compressor of a given size?

10. The refrigerating capacity of a small cold storage room is 5 tons per day of 24 hours. What displacement would be necessary in a small vertical single-acting ammonia compressor to produce the above refrigeration in 8 hours operating between the pressures of 15 lbs. gauge suction pressure and 170 lbs. gauge condenser pressure when the volumetric efficiency is 65 per cent?

CHAPTER VII.

THE ABSORPTION AND OTHER REFRIGERATION SYSTEMS.

The Absorption Refrigeration System.—The absorption system is constructed principally in three different types at present. These are the tubular, double pipe, and atmospheric types. Formerly, a shell and coil type of machine was built. The tubular type of construction is shown by Fig. 47 of Chapter V. The double pipe type of machine is illustrated by Fig. 48 of Chapter V and Fig. 78 of this Chapter. The atmospheric construction is indicated by Fig. 49 of Chapter V.

The type of construction to be used in any case depends upon the conditions in the plant. Thus the nature of the cooling water is a determining factor. If the cooling water is taken from wells, it may contain mineral matter which will deposit scale or sediment on the cooling surfaces. In this case, it is advisable to employ the atmospheric type of machine. Similarly, if the water is taken from rivers, lakes, or canals, it may contain mud, leaves, bits of wood, and other suspended matter, and in this case it is desirable to use the atmospheric type of plant. But when the cooling water is free from foreign matter and scale-forming solids, the double pipe may be used efficiently. Also, it is generally advantageous to use the atmospheric type when the initial temperature of the water is fairly high.

Generators.—Generators are made in two types—the shell and coil, and the multi-tubular types. The shell and coil type consist of horizontal shells containing a continuously welded coil for the steam. This type of construction is shown by Figs. 79, 80. The multi-tubular type consists of a shell which contains a number of pipes in a horizontal position. These pipes are connected to a steam compartment at the end of the generator, as shown by Figs. 79, 80.

Analyzer.—This piece of apparatus is generally constructed by means of a horizontal or vertical shell which contains a series of trays. The incoming strong aqua flows over these trays in counter-current to the outgoing gases, thereby promoting the proper heat interchange. This is shown by Fig. 81.

Absorbers.—Absorbers are constructed generally in the tubular, double-pipe and atmospheric types. The tubular type consists of a horizontal shell with tube sheets at the ends. Tubes placed on very close centers are expanded into the heads. Heavy cast iron water heads are bolted to the tube heads, which are designed in such a manner as to cause the water to make a number of passes through the tubes before leaving the absorber. This method distributes the cooling water evenly through each tube, giving it a high velocity, consequently high heat transmission is obtained.

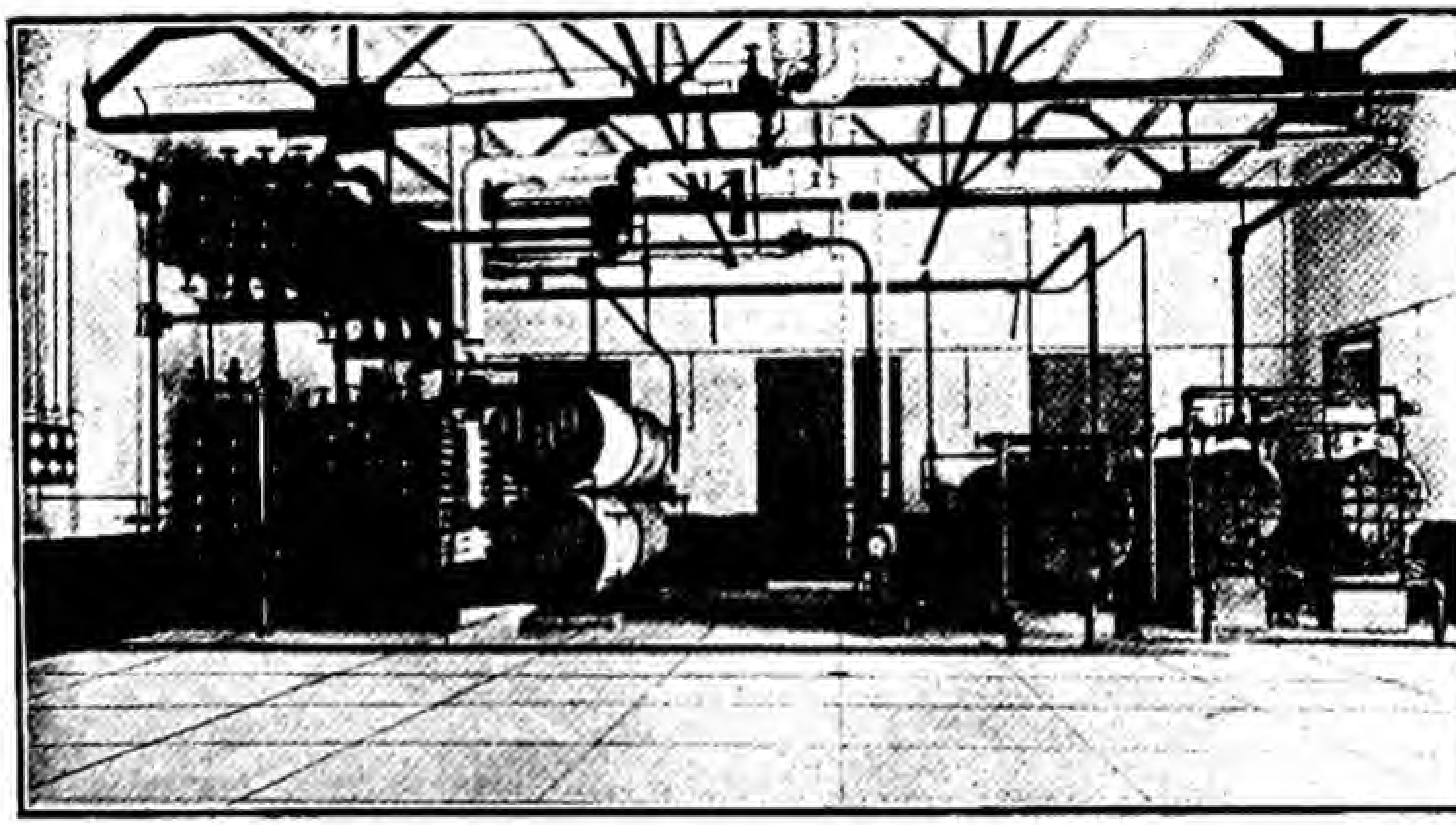


Fig. 78.—Vogt Double-Pipe Absorption Machine.

The weak aqua ammonia and gas are admitted to the bottom of the shell, into a mixing device.

The double-pipe absorber consists of 2-in. and 3-in. pipe, with stands, headers, valves, fittings, special gas and weak aqua ammonia injection fitting, and purging drum. The gas from the evaporator and the weak aqua ammonia are injected into the annular space of the coil at the bottom, where it is thoroughly mixed, passing through the various pipes in a thin film, which is always in contact with the cooling water pipe surface, causing the gas to be quickly absorbed by the weak aqua ammonia. The strong solution overflows at the top of the coil into the strong aqua ammonia tank. The cooling water enters the inner pipe at the top and leaves at the bottom of the coil, in countercurrent with the ammonia.

The atmospheric absorber is made of 2-in. pipe, 20 ft. long and has stands, headers, valves, fittings, and special injector fitting. The operation is similar to the double pipe type, the gas and the weak aqua ammonia entering the bottom pipe, overflowing as a strong solution

from the top pipe into a strong aqua tank. The cooling water is distributed over the outer surface of the pipe, in counter-current with the ammonia.

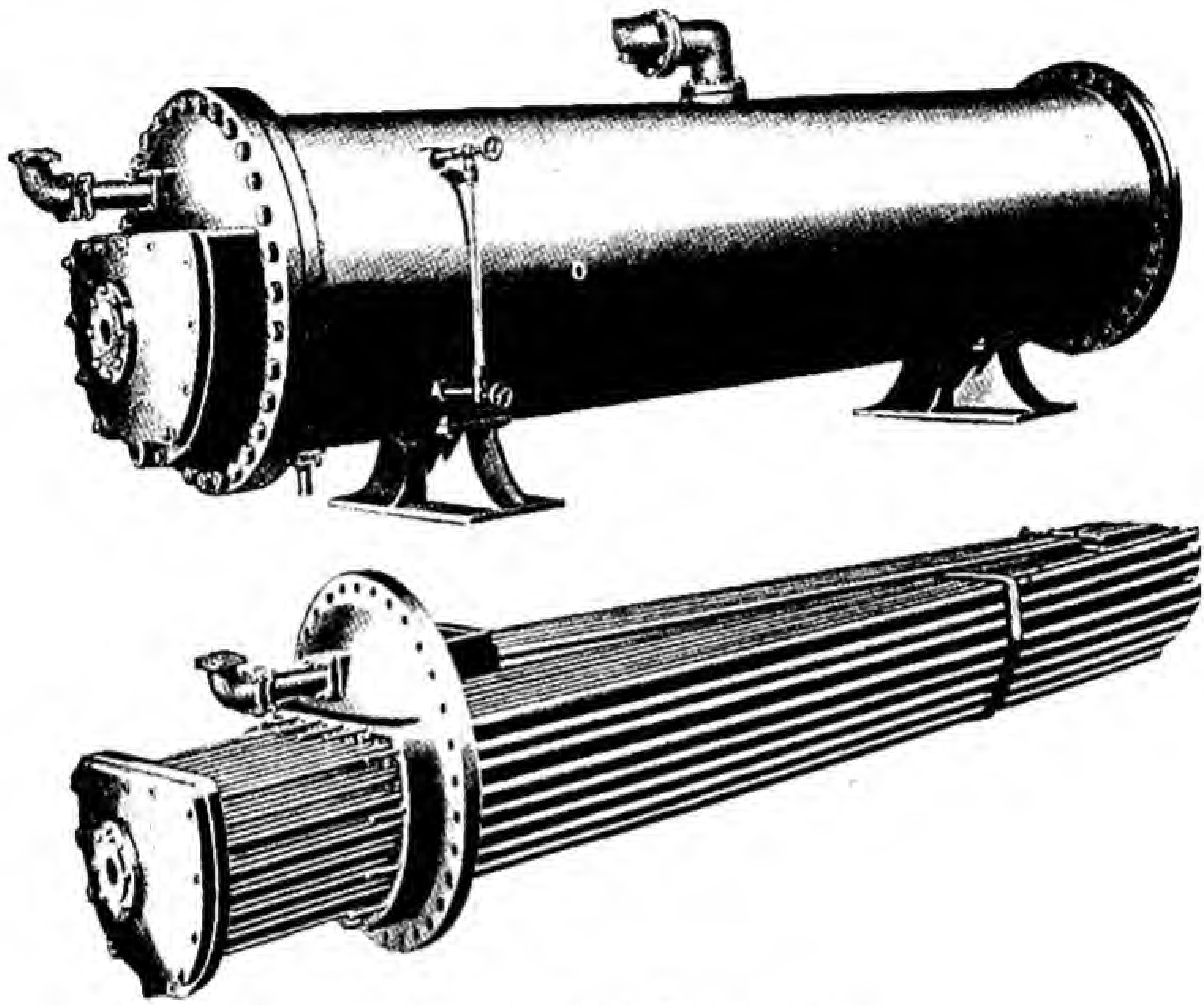


Fig. 79.—Vogt Generator.

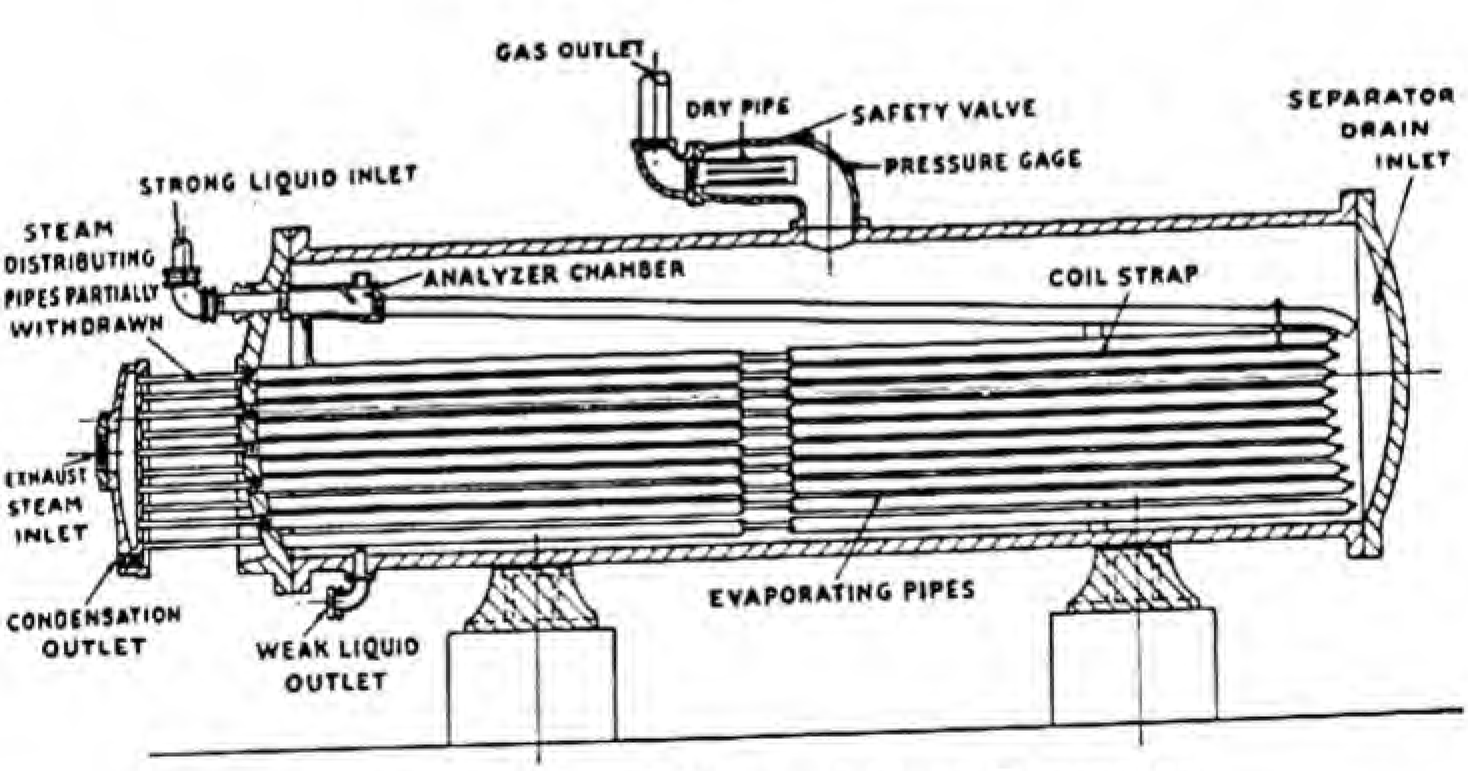


Fig. 80.—Sectional View of Vogt Generator.

Rectifiers.—Rectifiers or dehydraters are constructed in the double-pipe, atmospheric, and tubular types.

The double-pipe rectifiers are made of 1½-in. and 3-in. ammonia pipe, and are provided with all necessary fittings, valves, headers,

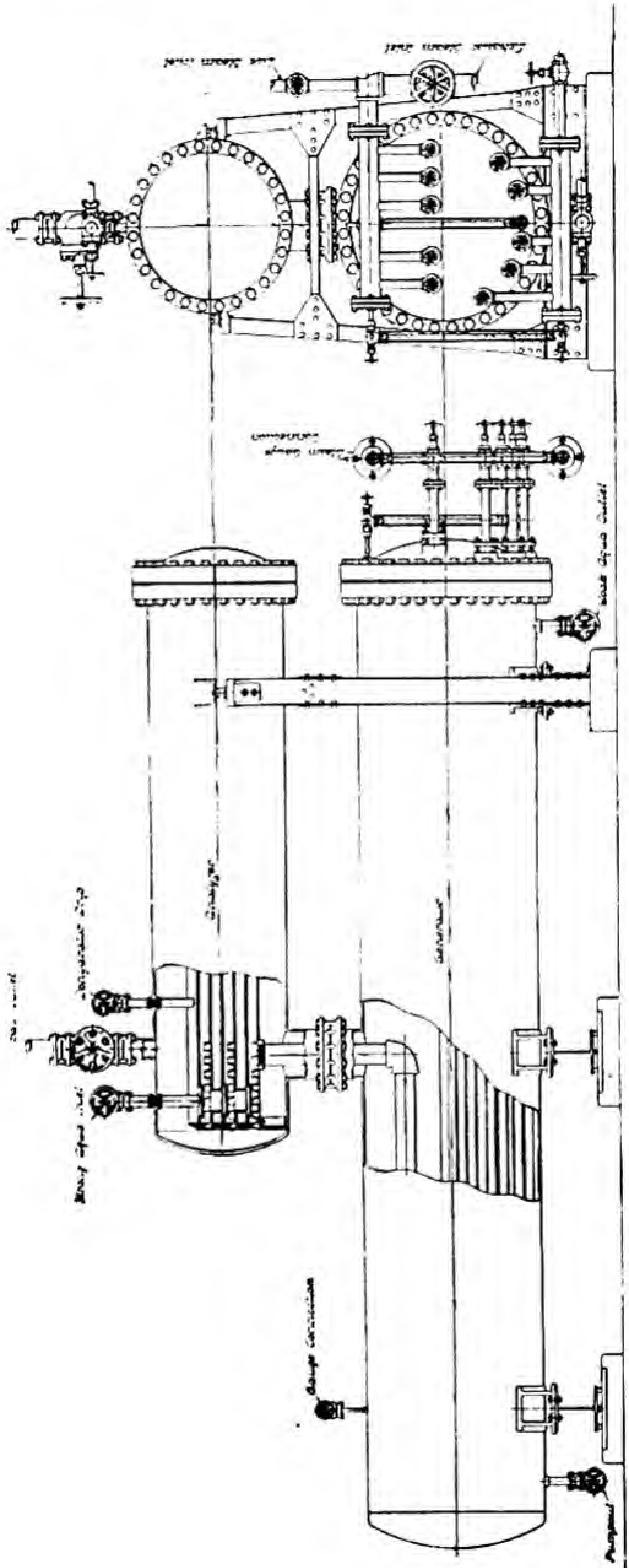


Fig. 81.—York Generator and Analyzer.

stands, and a drip trap. The combined gas and water vapors from the generator enter the annular space of the coil at the top, and leave it at the bottom, where it enters the drip trap, which separates the condensed water vapor from the ammonia gas, and returns it to the generator, while the anhydrous ammonia gas passes on to the ammonia condenser, or the drip may be drained off at intervals. The cooling water is circulated in counter-current with the ammonia, entering the inner pipe of the coil at the bottom and leaving it at the top.

The atmospheric rectifier is a vertical straight coil, provided with all the fittings, valves, headers, stands, cooling water distributing devices, and a drip trap. The gas travels in the same manner as in the double-pipe type, while the water is showered over the pipe on the outside.

The tubular rectifier consists of a series of shells which contain tubes. These tubes extend between the heads. The cooling water or strong aqua ammonia from the absorber passes through the tubes in counter-current to the gases from the generator. The gases are outside of the tubes and inside of shell. This type of rectifier is shown by Fig. 47 of Chapter V.

Exchangers.—Exchangers are either of the double-pipe, shell-and-coil, or the tubular types.

The counter-current double-pipe exchanger is a very efficient exchanger, and is made of either 1¼-in., 2-in. or 3-in. pipes, 20 ft. long. The coils are arranged so that both the strong and weak aqua ammonia travel at a high velocity, thereby gaining a better heat transmission than could be obtained in any other type of exchanger. This causes the strong aqua ammonia to enter the generator at a temperature very little lower than the temperature of the generator.

The shell-and-coil type exchanger consists of a welded steel shell, with iron heads. The shell contains a series of spiral coils made of extra heavy 1¼-in. pipe, through which the weak aqua ammonia is circulated, the strong aqua ammonia passing through the shell. As the velocity of the ammonia is low, the heat transmission is not as good as can be expected in the double-pipe type. Thus, unless floor space is limited, the double-pipe type is used on account of its greater efficiency of heat transmission.

The construction of the tubular exchanger is very similar to that of the tubular rectifier.

Weak Aqua Coolers.—This apparatus is constructed in the double-pipe, atmospheric, and tubular types.

The double-pipe weak aqua cooler is made of 1¼-in. and 2-in. pipe, 20 ft. long with stands, headers, valves, and fittings. The weak aqua ammonia from the exchanger enters the annular space at the bottom

of the coil, leaving at the top, while the cooling water enters the inner pipe at the top and passes through the coil in counter-current with the ammonia, causing the weak aqua ammonia to leave the coil at a temperature within a few degrees of the temperature of the cooling water.

The atmospheric weak aqua cooler is a simple coil 2-in. pipe 20 ft. long, and has stands, headers, valves, fittings, and water distributing devices. The operation is similar to the double-pipe type, the weak aqua ammonia entering the bottom pipe of the coil and leaving it at the top, while the water is distributed over the outer surface of the pipe, in counter-current with the flow of the ammonia.

The construction of the tubular weak aqua cooler is identical with that of the tubular rectifier.

Aqua Ammonia Pump.—Aqua ammonia pumps are of the direct acting piston type. They may be either steam or belt driven.

An extra deep stuffing-box is provided on the ammonia end of the pump, with a lantern gland, from which a connection is made to the suction side of the cylinder, causing any leakage from the cylinder through the first part of the stuffing-box to be carried back into the suction side of the cylinder, thereby allowing the pump to operate without leakage or drip from the stuffing-box.

The piston rods are made of steel in two sections, thereby allowing the ammonia piston rod to be easily replaced, without interfering with or removing the steam portion of the rod, the rods being connected by means of a forged steel coupling.

The belt driven aqua pump has a shaft, belt-wheel, and necessary gears for speed reduction, instead of the steam cylinder.

Fig. 82 shows the general arrangement of the York double-pipe absorption refrigerating machine.

Fig. 83 shows the arrangement of the Vogt shell-and-tube absorption machine.

The Intermittent Absorption Machine.—The intermittent or alternating absorption machine is sometimes constructed in small sizes. This machine usually has no working parts and depends upon the application of heat to produce the necessary distillation. The still or generator is used alternately as a generator and absorber.

When operating as a generator, part of the ammonia is driven out by means of heat produced by either a gas flame or electric current. The heat is applied slowly and very little water vapor is driven out with the ammonia gas.

The ammonia gas is condensed and stored in a liquid receiver. The liquid ammonia is then automatically fed to the evaporating unit, and is evaporated into a vapor, producing the required refrigerating effect.

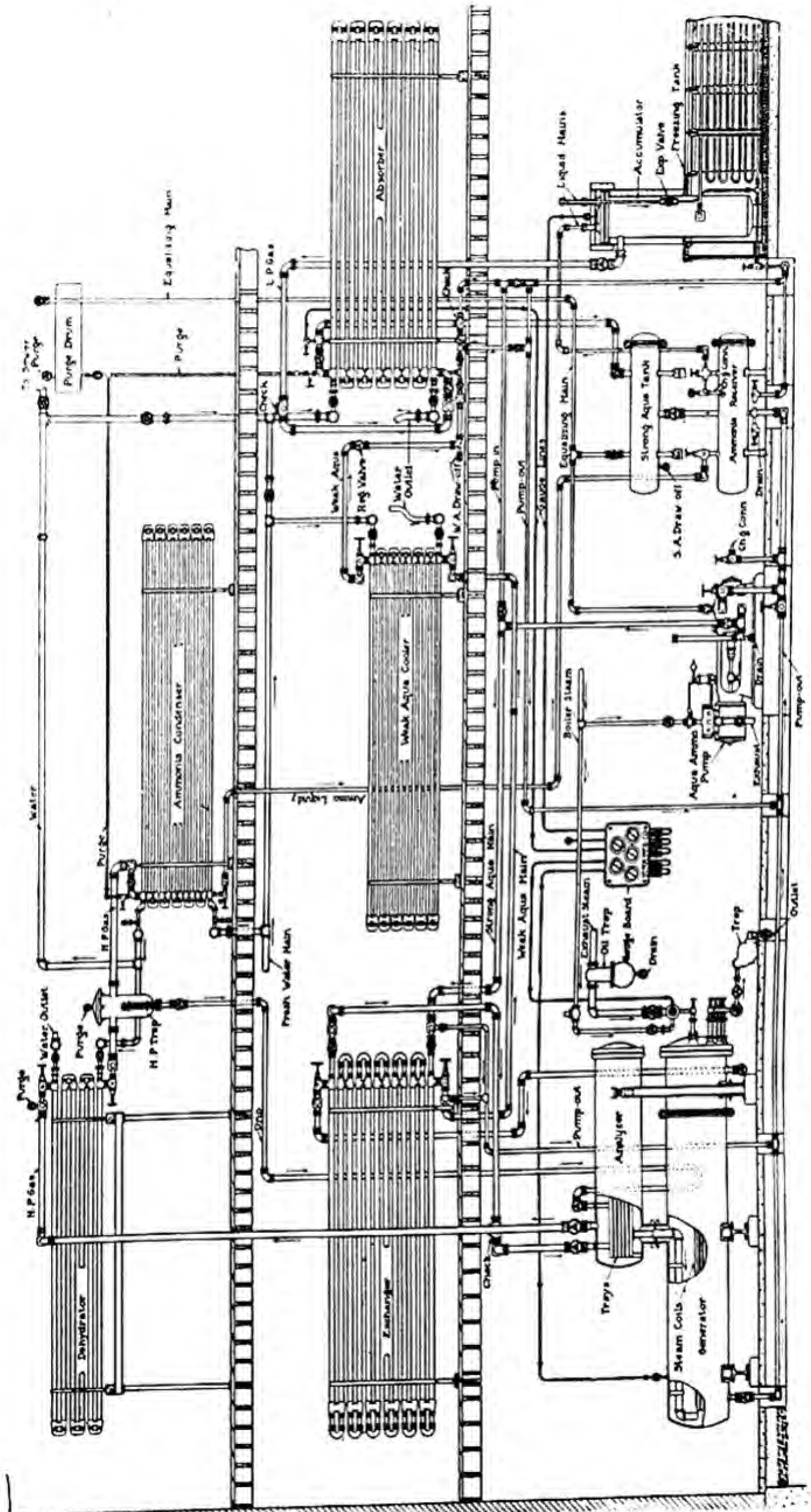


Fig. 82.—York Double-Pipe Absorption Machine.

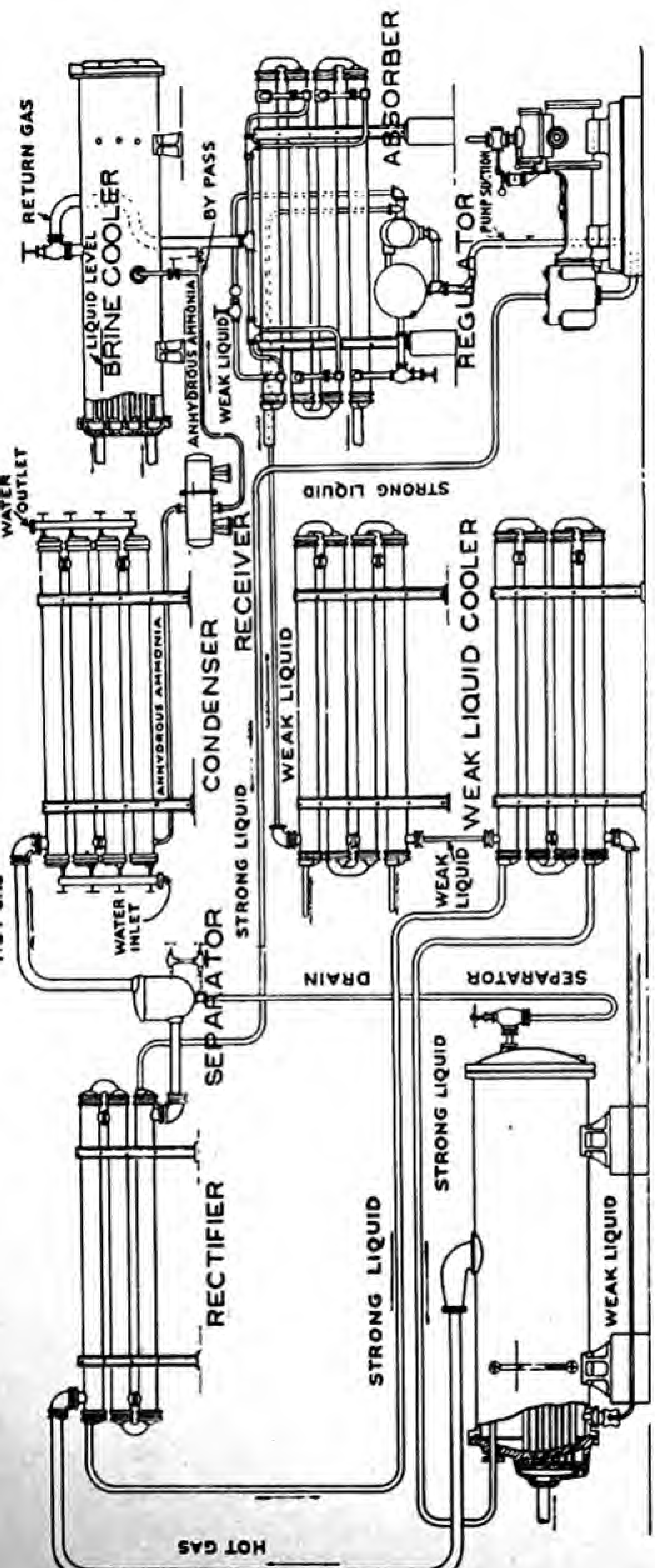


Fig. 83.—Vogt Shell-and-Tube Absorption Machine.

After the ammonia has been partly boiled out of the solution in the generator, the source of heat is withdrawn, cooling water is admitted, and the generator now becomes an absorber, and is then ready to absorb the vapor of ammonia produced by the evaporating unit. The re-absorption of the gas by the weak aqua in absorber changes it again to strong aqua, which is now ready to be reheated and re-distilled.

Refrigeration is interrupted during the distilling process which usually requires two to three hours per day of twenty-four. The remainder of the time is devoted to active evaporation of the liquid ammonia in the refrigerating unit. The alternations may be controlled by automatic devices.

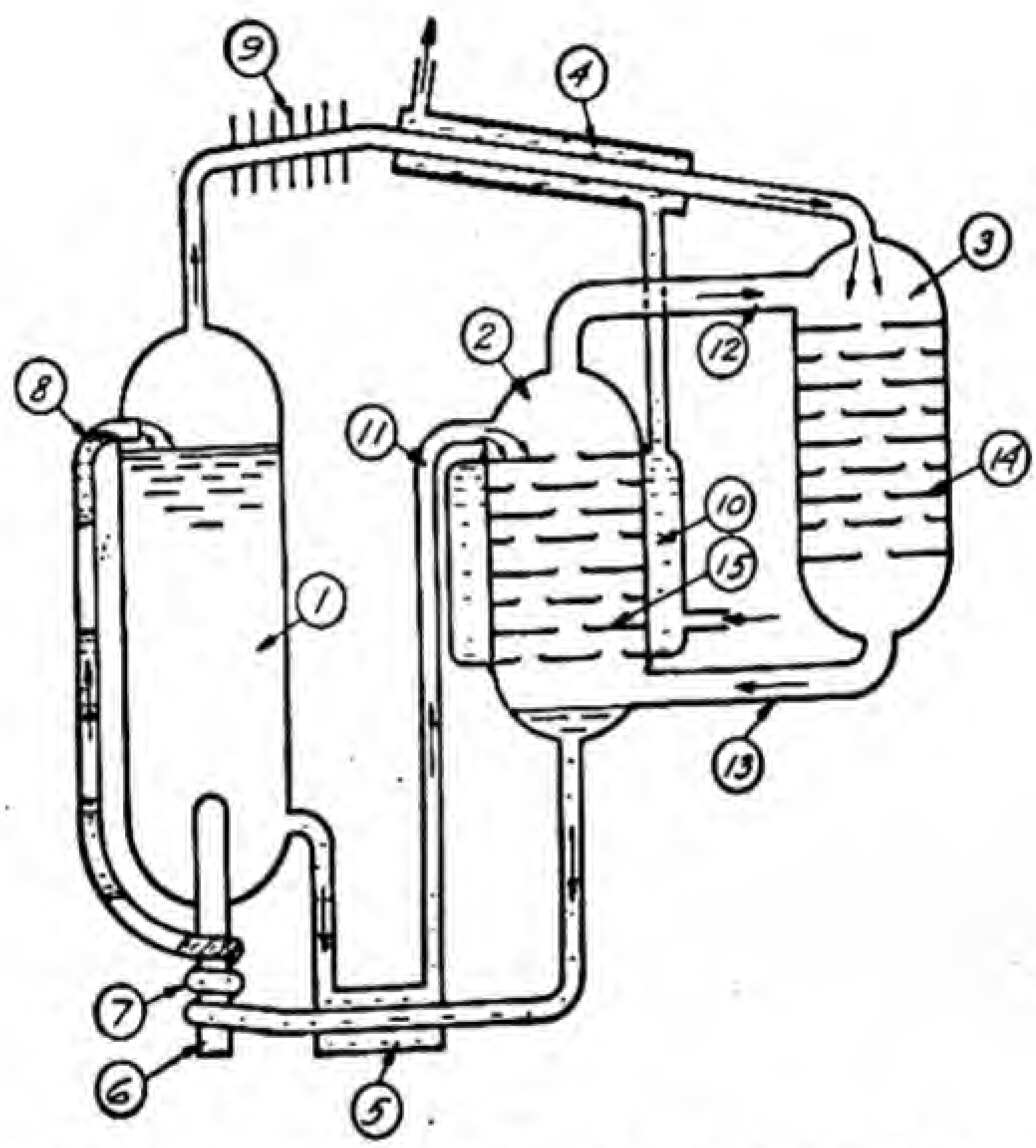


Fig. 84.—Diagrammatic Sketch of Swedish System.

Electrolux Refrigerator.—The commonly known type of ammonia absorption refrigerating machine requires a reduction of pressure between the condenser and the evaporator, and an increase of pressure between the absorber and generator. In the Electrolux refrigerating system, the equivalent to decreased and increased pressures is obtained by the introduction of a gas such as hydrogen. The presence of this gas produces virtually a drop in vapor pressure of the refrigerating agent, due to the fact that as the gases mix, the partial pressure of the refrigerating agent falls, thereby making it possible for the refrigerating agent to evaporate at a low pressure and temperature.

A diagrammatic sketch of this refrigerating system is shown in Fig. 84, which shows that the apparatus is composed of three principal parts, namely, generator, evaporator, and absorber. The following numbers refer to the respective parts: 1, generator; 2, absorber; 3, evaporator; 4, condenser; 5, temperature exchanger; 6, heating medi-

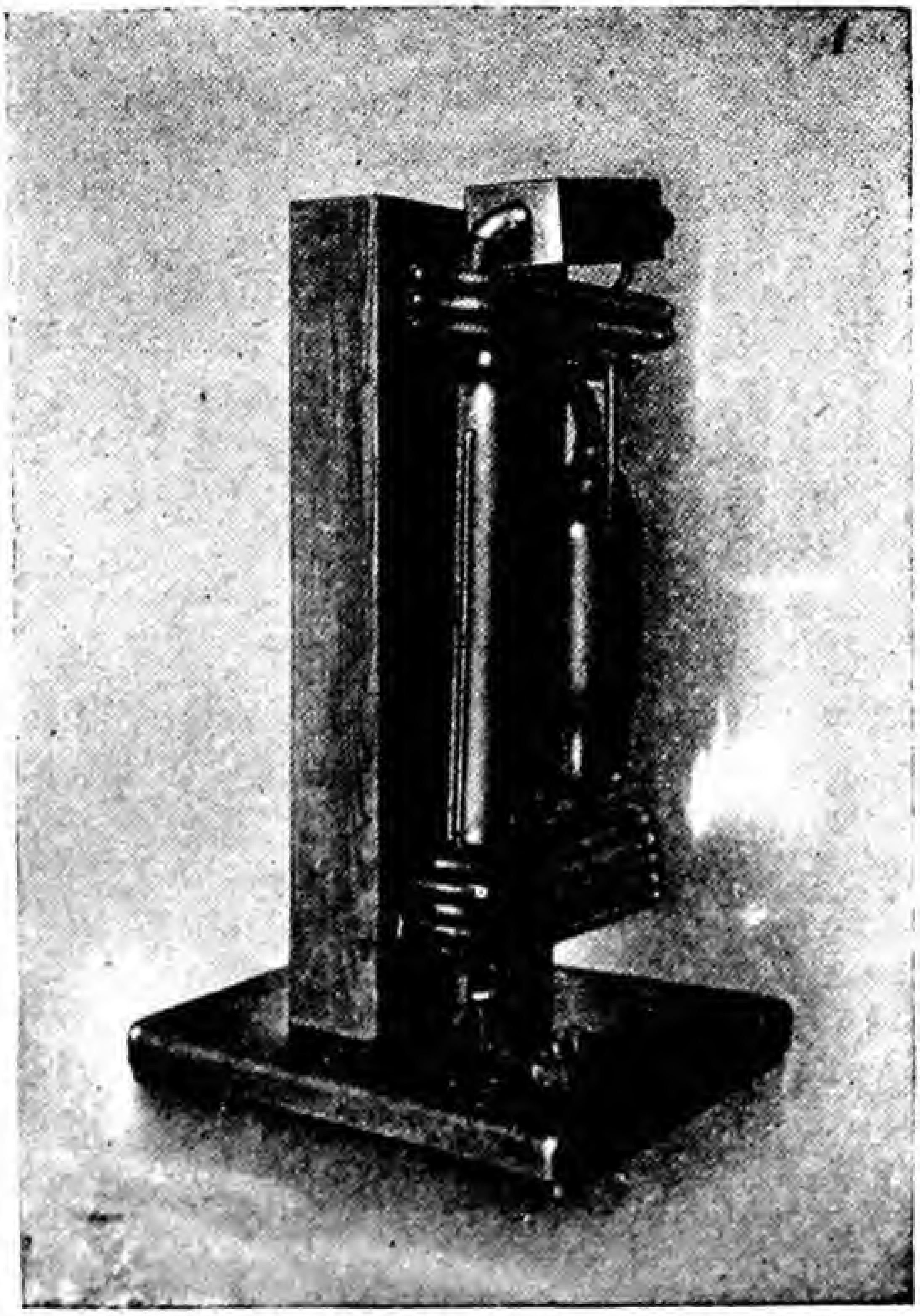


Fig. 85.—View Showing Simplicity of Construction.

um; 7, thermo-syphon; 8, strong liquor inlet; 9, liquefier; 10, cooling jacket; 11, weak liquor inlet; 12, hydrogen inlet; 13, mixed gas outlet; 14, discs in evaporator; 15, discs in absorber. When the generator (1) is heated, the ammonia which has been previously dissolved in the water, is evaporated or distilled from the solution. This gas passes through the liquefier line and is condensed in the condenser (4). The liquefied ammonia flows from the condenser to the evaporator (3), where it is met by the hydrogen gas, which is continuously transmitted

from the absorber (2) by means of pipe connections (12). The liquefied ammonia flows downward over a number of discs (14) in the evaporator, where it evaporates, diffusing itself in the hydrogen. This evaporation and mixing goes on until the ammonia vapor has reached the partial pressure in the mixture of gases which corresponds to the existing conditions of temperature and pressure in the evaporator.

This gas mixture consists partly of hydrogen and partly of ammonia, and has a higher specific gravity than that of hydrogen, and therefore sinks to the bottom of the evaporator (3). From the bottom of the evaporator, the gas mixture flows through pipe (13) to the absorber (2) where it is met by a shower of water practically free of ammonia, coming through pipe (11) and passing over discs (15) in the absorber. The water absorbs the ammonia allowing the hydrogen to return to the evaporator through connection (12). The strong ammonia liquor from the bottom of the absorber is led into the top of the generator through pipe (7, 8), which acts as a thermo-syphon, due to the application of heat. Pipe (5) is placed inside of pipe (11) in order to act as a heat exchanger.

The assembled view of this apparatus is shown in Fig. 85, which illustrates the simplicity of construction.

Carbon Dioxide Compression System.—The carbon dioxide system is quite similar in general construction to that of the ammonia compression system. The principal difference of the design of the system is due to the fact that the condenser pressure is quite high, ranging from 700 to 1,000 lbs. per sq. in. This high pressure necessitates the use of heavier parts, deeper stuffing-boxes, etc.

Carbon dioxide compressors are constructed in both the horizontal double-acting, and the vertical single-acting types. The smaller size machines are constructed in the vertical single-acting type, while larger sizes up to 200 tons refrigerating capacity are constructed in the horizontal double-acting type. A horizontal double-acting machine is illustrated by Fig. 86. As previously indicated, the compressor cylinders are comparatively small as compared with those of the ammonia compression system and others.

The compressor rod stuffing-box is similar in general construction to the stuffing-box of the ammonia compressor. A large amount of good packing must be used, and the packing must be well lubricated and properly adjusted in order to withstand the high pressure.

Fig. 87 shows a Frick vertical twin cylinder carbon dioxide compressor.

Carbon Dioxide Condensers.—The condensers used in the carbon dioxide compression system are generally the double-pipe, the atmospheric, and shell-and-tube types.

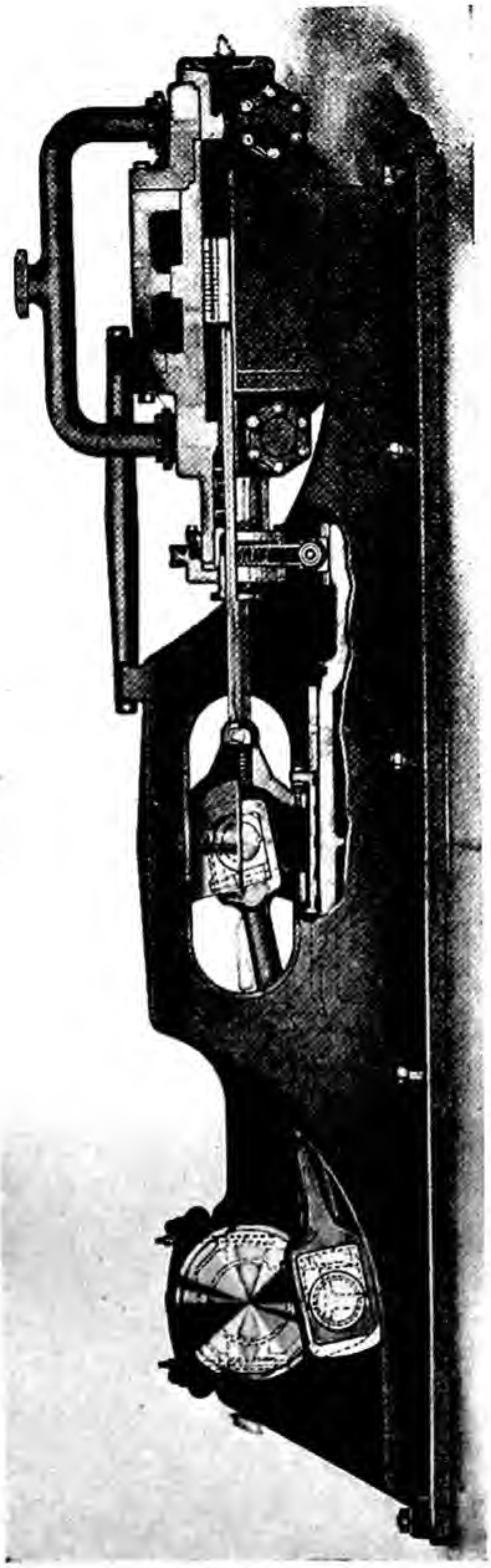


Fig. 86.—Cross-Section American Carbonic Machinery Compressor.

In general construction, the double-pipe condenser is similar to that of the ammonia compression system. The outer pipes are connected to the double-pipe fittings by means of flanged joints. The fittings for the condenser are generally made of semi-steel and are extra heavy. A special form of packing, constructed of hard rubber or metal, is used to prevent leakage. The fittings may be eliminated also, by means of welding.

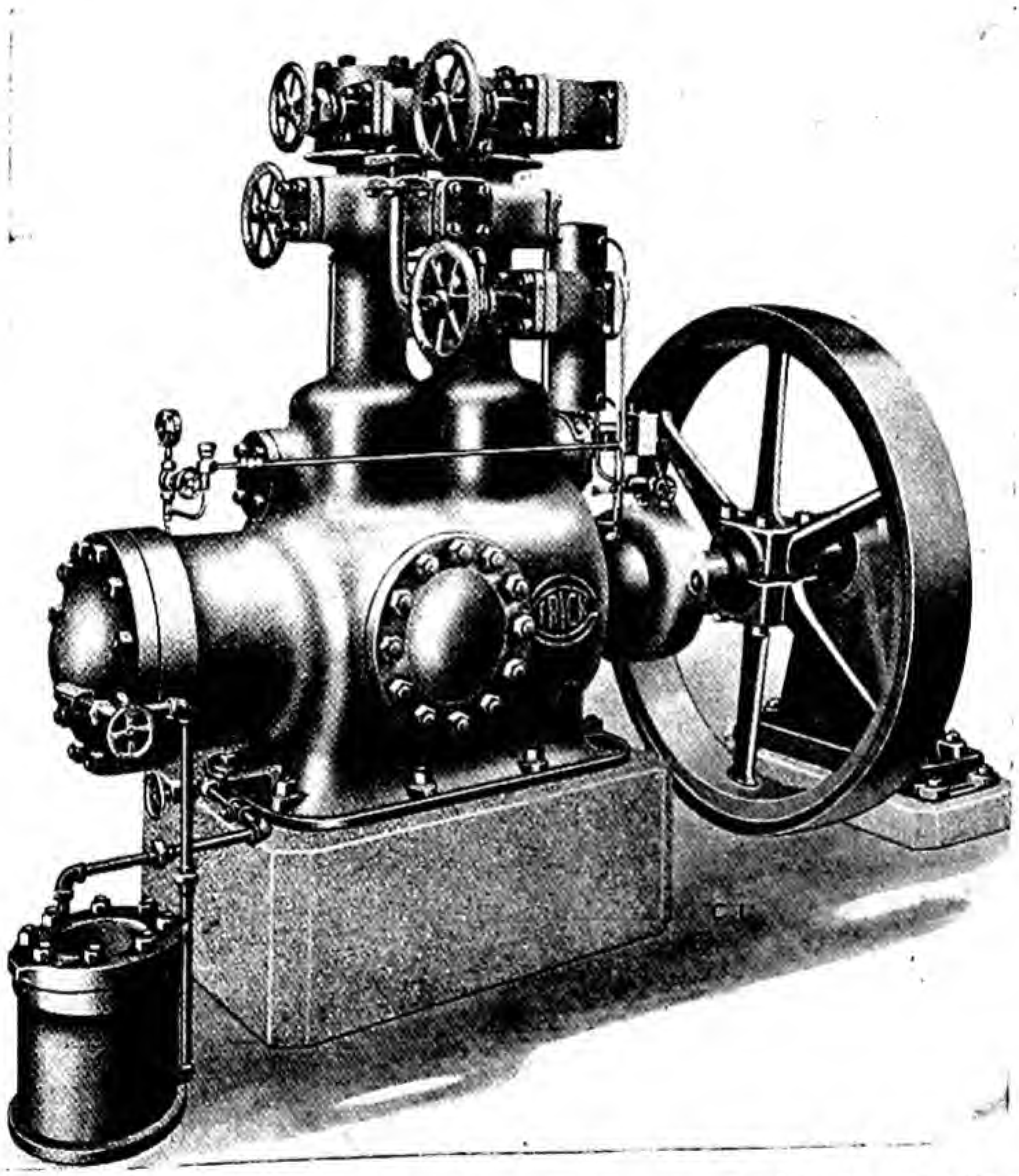


Fig. 87.—Frick Vertical Single-Acting Compressor Carbon Dioxide Type.

The atmospheric type condenser is generally made up from continuously welded pipe with flanged fittings for connecting the ends of the coils.

Shell-and-tube carbon dioxide condensers are usually of the closed multi-pass type.

Valves and Fittings.—Valves and fittings for the carbon dioxide compression system are usually made of semi-steel and are extra heavy. The smaller valves and flanges are sometimes made from steel forgings. All valves and fittings are provided with companion flanges, and all pipe connections are made by means of flanged unions. Screwed couplings and screwed fittings are not used in the system at all. The flanged joints are maintained tight by means of suitable gaskets.

Application of the Carbon Dioxide System.—In general, the carbon dioxide system of refrigeration may be used for any of the industrial applications of refrigeration. Some of the relative advantages and disadvantages of the use of carbon dioxide were mentioned in Chapter III.

The carbon dioxide system is particularly well adapted to the production of extremely low temperatures. This is due to the fact that as low a temperature as −110° F. may be obtained by reducing the suction pressure to that of the atmosphere. The advantage of keeping the suction pressure above the atmosphere is obvious. Carbon dioxide as a refrigerant has the further advantage of being fairly low in cost. It is well adapted for use in hotels, on board ship, and other places where the possible escape of the refrigerant might be dangerous to human life.

Solid Carbon Dioxide.—A recent letter circular issued by the Bureau of Standards of the U. S. Department of Commerce contains data on solid carbon dioxide giving density, temperature, vapor pressure, latent heat of sublimation, latent heat of fusion, specific heats low temperatures, and uses.

Solid carbon dioxide compressed into cakes is being manufactured for use as a refrigerant and sold under several trade names, such as "Dry Ice," "Carbonice," etc. These cakes of compressed solid carbon dioxide closely resemble packed snow in appearance and have a temperature of −109° F. (−78.5° C.) or lower. Carbon dioxide at room temperatures and atmospheric pressure is a colorless, odorless gas. It occurs in the atmosphere to the extent of about 0.03 per cent by weight, is a product of combustion and respiration, and a by-product of fermentation and of many chemical processes. It can exist as a solid at atmospheric pressure only because of its very low temperature. The cakes are made by compressing in a mold carbon dioxide snow produced by expanding liquid carbon dioxide at a low temperature from a high pressure to atmospheric pressure. In the expansion, part of the liquid is changed to a solid in the form of snow; the rest becomes a gas which is returned to the compressor for recompression and the making of more snow.

Carbon dioxide is different from water and most other substances in that it cannot exist as a liquid at atmospheric pressure (14.7 lbs. per

sq. in.).[1] Only when the pressure is equal to or greater than 75.1 lbs. per sq. in. (5.1 normal atmospheres) and its temperature −70° F. (−56.6° C.) or higher (the "triple point" pressure and temperature) can carbon dioxide exist as a liquid. Hence instead of melting to a liquid as ice does, solid carbon dioxide sublimes, that is, it passes directly from the solid to the gaseous state. This is one of the great advantages of solid carbon dioxide when used as a refrigerant. It does not wet spaces, packages, and materials refrigerated with it, and all the inconveniences due to the water from melting ice are avoided.

Density.—The density of the commercial product depends upon the pressures applied in compressing the loose solid into cakes, and possibly on the manner of compressing it. A sample tested at the Bureau of Standards weighed 79 lbs. per cu. ft. or 1.27 grams per cc., which may be compared with 57 lbs. per cu. ft. or 0.92 gram per cc. for ordinary ice. Crystalline carbon dioxide made by freezing liquid carbon dioxide weighs about 96 lbs. per cu. ft., or 1.53 grams per cc., (International Critical Tables, Vol. I, p. 112 and Vol. III, p. 43).

Temperature.—The temperature of solid carbon dioxide surrounded by pure, gaseous carbon dioxide at a pressure of one normal atmosphere is −109° F. or −78° C. (International Critical Tables, Vol. III, p. 207). In contact with air, its temperature is lower because the partial pressure of carbon dioxide gas is less. In contact with quiescent dry air a temperature of −114° F. has been observed and in air currents even lower temperatures are observed.

Vapor Pressure.—Pure solid carbon dioxide enclosed in a container in contact with its own vapor exerts a pressure, known as the vapor pressure, which varies with the temperature of the solid. The following table, which is an unpublished correlation made at the Bureau of Standards of various data, shows the variation of the vapor pressure of carbon dioxide with temperature.

The normal atmosphere is defined as a pressure exerted by a column of mercury 76 cm. high under standard conditions and is very closely equal to a pressure of 14.7 lbs. per sq. in.

Latent Heat of Sublimation.—In passing from the solid to the gaseous state at atmospheric pressure carbon dioxide takes up 248 Btu. of heat energy per pound, or 138 calories per gram (International Critical Tables, Vol. V, p. 138). A Btu. (British thermal unit) is by definition the quantity of heat energy required to raise the temperature of one pound of water 1° F.

[1]An excellent discussion of this subject was written by C. H. Meyers for *Ice and Refrigeration*. See "Carbon Dioxide in the Solid, Liquid and Vapor States," Vol. 76, pp. 535-37, June, 1929.

Latent Heat of Fusion.—In passing from the solid to the liquid state at its "triple point" (−70° F. or −56.6° C.) carbon dioxide takes up 82 Btu. of heat energy per pound, or 45.3 calories per gram (International Critical Tables, Vol. V, p. 131).

TABLE 55.—VARIATION OF VAPOR PRESSURE OF CARBON DIOXIDE WITH TEMPERATURE.

Temperature	Absolute Pressure		Temperature
° F.	lbs./sq. in.	Normal Atmospheres	° C.
−69.8	75.1	5.11	−56.6
−75	61.7	4.20	−59.4
−80	50.8	3.46	−62.2
−85	41.6	2.83	−65.0
−90	33.9	2.31	−67.8
−95	27.5	1.87	−70.6
−100	22.2	1.51	−73.3
−105	17.8	1.21	−76.1
−110	14.2	0.97	−78.9
−120	8.9	0.61	−84.4
−130	5.4	0.37	−90.0
−140	3.2	0.22	−95.6

Specific Heats of Solid and Gaseous Carbon Dioxide at Low Temperatures.—The specific heat of solid carbon dioxide at −109° F. (−78.5° C.) is 0.31 Btu. per lb. per ° F. or calorie per gram per ° C.; that is, in order to raise or lower the temperature of one pound of solid carbon dioxide 1° F. at −109° F., 0.31 Btu. of heat energy has to be added to or taken from the solid, accordingly as its temperature is to be raised or lowered (International Critical Tables, Vol. V, p. 95). In the temperature interval between −109° F. (−78.5° C.) and +32° F. (0° C.) the mean specific heat of the vapor is about 0.19 Btu. per lb. per ° F. or calorie per gram per ° C. Therefore one pound of carbon dioxide vapor after subliming from the solid will absorb 0.19 Btu. for each degree rise in temperature between −109° F. and +32° F. (International Critical Tables, Vol. V, p. 80).

Refrigerating Effect.—Besides the refrigerating effect due to the change of state, there is the additional refrigerating effect of 27 Btu. per lb. at 32° F. (15 calories per gram at 0° C.) equal to the amount of heat which the cold carbon dioxide vapor at −109° F. (−78.5° C.), after subliming from the solid, absorbs in being warmed to 32° F. (0° C.). Hence one pound of carbon dioxide absorbs 275 Btu., or 153 calories per gram, in changing from a solid at −109° F. (−78.5° C.) to a gas at 32° F. (0° C.). This is approximately equal to twice the amount of heat, 144 Btu. per lb. or 79.6 calories per gram absorbed by ice on melting at 32° F. (0° C.); and, as this is often expressed, one

pound of solid carbon dioxide has approximately the same refrigerating effect at 32° F. as two pounds of ice.

Mixtures of ice and salt are used to produce temperatures below 32° F. and to refrigerate spaces at temperatures below those obtainable with ice alone. The heat absorbed by a pound of a mixture of ice and salt when the ice melts and the salt dissolves—its refrigerating effect —is smaller than the heat absorbed by one pound of ice, inasmuch as a pound of the mixture contains less than a pound of ice, the salt affecting the result only to a comparatively small extent. Moreover, the latent heat of fusion of ice is smaller at lower temperatures than it is at 32° F., and some of the ice is melted in cooling the mixture of ice and salt to the reduced temperature. These effects lower somewhat the amount of refrigeration which can be obtained from ice when it is used with salt. Hence, the ratio of the refrigerating effect of solid carbon dioxide to that of ice used with salt is greater than the ratio of their refrigerating effects when ice is used alone.

As solid carbon dioxide is ordinarily used as a refrigerant, the cold carbon dioxide gas as it sublimes from the solid displaces from the space to be refrigerated first the air and then the warm carbon dioxide gas. Gaseous carbon dioxide is a better heat insulator than air, the ratio of the heat conducted by carbon dioxide to that conducted by air under the same conditions at 32° F. being 0.6. The heat, however, that passes from a warm exterior into a refrigerated space depends, among other things, upon the insulating properties and upon the thickness of the separating walls as well as upon the thermal conductivity of the gas inside. The better the heat insulation of the separating walls, the relatively less important is the thermal conductivity of the gas inside.

Uses.—Solid carbon dioxide is used most extensively for refrigerating ice cream and other frozen goods in transit. It is also used for refrigerating shipments of other perishable commodities. In laboratories it is used to some extent for the production and maintenance of low temperatures for testing and experimental work.

Ethyl Chloride System.—Such low pressure refrigerants as ethyl chloride, methyl chloride, and others, are coming into use more and more for the production of refrigeration in small refrigerating plants such as those for residences, meat markets, grocery stores, ships, soda fountains, restaurants, etc.

Ethyl chloride may be taken as an example of this type of plant. In this system, the pressures are quite low, the suction pressure varying from 15 to 25 in. of vacuum and the condensing pressure varying from 6 to 12 lbs. per sq. in. gauge. In general, since the pressure of the re-

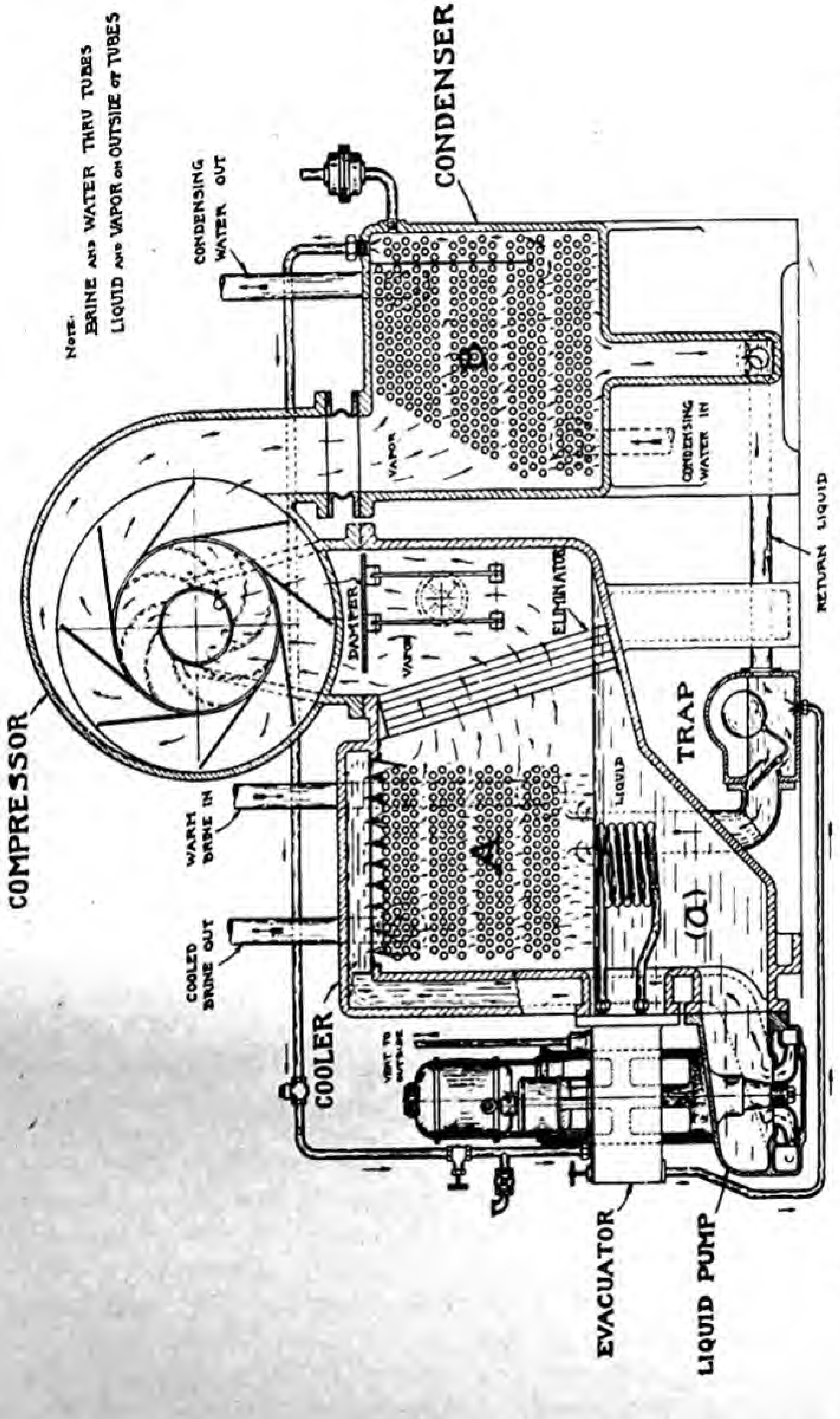


Fig. 88.—Diagrammatic Arrangement of Carrier Centrifugal Refrigeration System, 1927.

frigerant is quite low, in order to produce the desired refrigerating effect, a comparatively large volume of the vapor must be taken from the evaporator.

Centrifugal Refrigeration System.—The Carrier system of centrifugal refrigeration, which was the first of this type, operates on the compression system principle, using a multi-stage centrifugal compressor for compressing a special low-pressure refrigerant. Some of the properties of various refrigerants used by Carrier are given in Table 56.

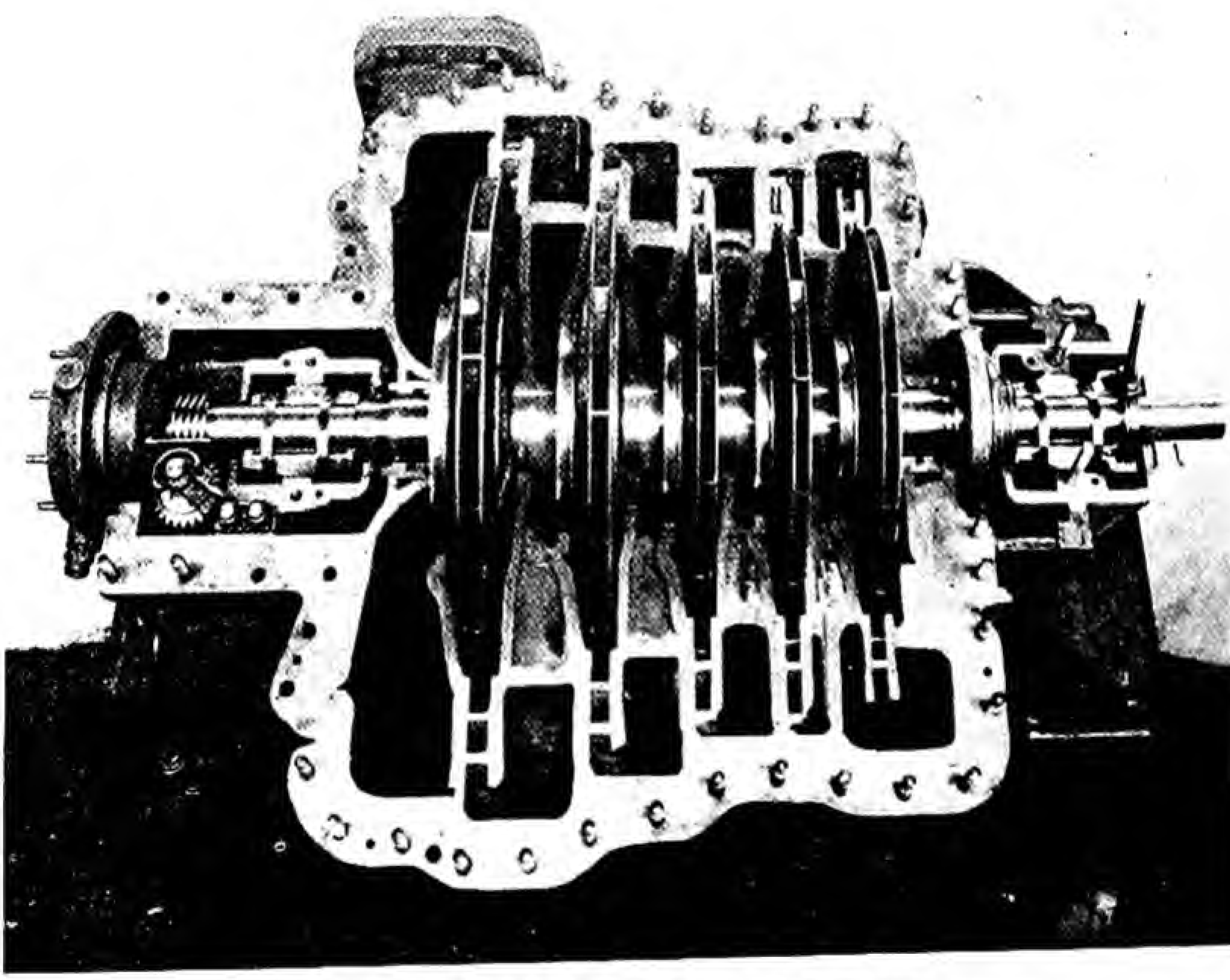


Fig. 89.—Carrier Centrifugal Compressor.

Fig. 88 shows the general arrangement of the Carrier centrifugal system (1927). Fig 89 is a view of a standard centrifugal compressor with the top half of the horizontally split shell removed.

Ice and Refrigeration for July, 1930, gives the following description of Carrier centrifugal systems installed at the Chicago Stadium:

In the Chicago Stadium two complete Carrier centrifugal refrigeration units comprise the present refrigerating equipment. Each of the two machines has a normal rating of 140 tons when cooling (Calcium)

brine to 14° F. Each is driven by individual 175-hp. Terry steam turbines operating at 115 lbs. pressure and 5 lbs. back pressure. These two units are installed in a secluded space—approximately 28 ft. x 15 ft. in the basement—occupying a total floor area of only 420 sq. ft.

Three major pieces of equipment operating as a compact unit comprise the centrifugal refrigeration machines. These are the evaporator or cooling unit, the centrifugal compressor, and the condenser.

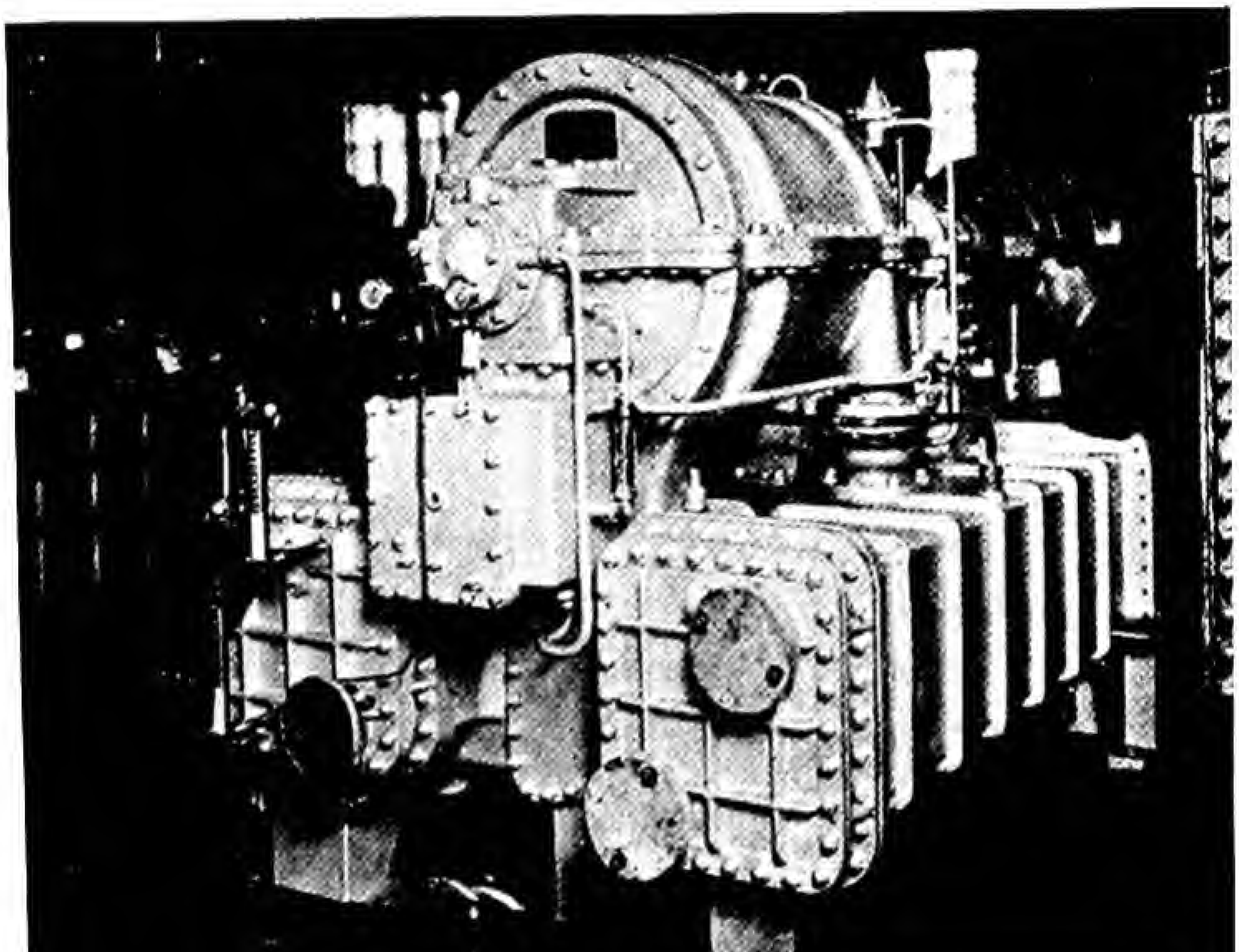


Fig. 90.—100-Ton Carrier Refrigeration Unit—Compressor End.

Fig. 90 shows a standard Carrier 100-ton gear-driven unit. The compressor is a simple multi-stage centrifugal unit and operates not at high pressures but at pressures below atmospheric. There is no sliding friction or gear compression—the only contact of moving parts being at the main shaft bearings. These are ring oiled and practically frictionless, requiring no attention whatever for indefinite periods.

The characteristics of the compressor are such that it cannot build up dangerously high pressures—even if its outlet is entirely closed, since the maximum pressure it can develop depends upon the rotative speed alone, and this is such that the closed outlet pressure would be but a few pounds gauge. The compressor operates at speeds in the neighborhood of 3,600 r.p.m. It may be driven by a directly connected squirrel-cage a.c. motor which has been developed for this purpose or,

it may be driven through gears by motors of lower speed; or, it may be driven by most any form of directly connected steam turbines.

The cooling of brine or water is accomplished within a shell and tube compartment. The liquid to be cooled is circulated through brass or bronze tubes. The liquid refrigerant is sprayed, or rather allowed to flow over the tubes within the shell and caused to evaporate under condition of twenty-five to twenty-eight inches of vacuum created on the intake side of the compressor.

Condensation of the refrigerant vapor is produced within the shell and tube condenser upon which the compressor rests as a base.

TABLE 56.—STANDARD TON DATA ON VARIOUS REFRIGERANTS USED BY CARRIER

Refrigerant	Dieline C2H2C12		Trieline C2HC13		Carrene CH2C12	
Subject	5°F	86°F	5°F	86°F	5°F	86°F
Absolute Pressure, lb./in.^{2}	.82	6.9	.16	1.72	1.29	10.3
Gage Pressure, lb./in.^{2}	−13.88	−7.8	−14.54	−12.98	−13.41	−4.4
Volume-Liquid, ft.^{3}/lb	.0127	....	.0109	....	.012	....
Volume-Vapor, ft.^{3}/lb.	63	8.5	240	25.2	48.3	6.8
Density-Liquid, lb./ft.^{3}	79	....	91.6	....	83	....
Density-Vapor, lb./ft.^{3}	.0159	.118	.00416	.0397	.0207	.147
Heat-Liquid. Btu./lb. above 0°F	1.35	23.2	1.17	20.1	1.7	29.2
Heat-Latent, Btu./lb	136	133	112.5	109.5	149	146
Heat-Vapor, Btu./lb.	137.35	156.2	113.67	129.6	150.7	175.2
Entropy-Liquid, Btu./lb. °F	.0029	.0425	.00252	.0368	.0037	.0535
Entropy-Evaporation, Btu./lb. °F	.293	.2435	.242	.201	.32	.267
Entropy-Vapor, Btu./lb. °F	.2959	.286	.2445	.2378	.3237	.3205
Specific heat of the liquid	.27		.233		.34	
Specific heat of the vapor C_p	.1625		.12		.154	
Specific heat of the vapor C_v	.1425		.105		.131	
Ratio C_p/C_v	1.14		1.14		1.18	
Specific gravity-liquid (water = 1)	1.27		1.47		1.33	
Specific gravity-vapor (air = 1)	3.36		4.55		3.00	
Critical Pressure, lb./in.^{2} (abs.)	* 800#/in.^{2}		....		* 1490	
Critical Temperature, °F	* 470°F		....		* 380	
Molecular Weight	96.9		131.35		84.9	
Melting Point, (1 atm.) °F	−70°F		−70°F		....	
Boiling Point, (1 atm.) °F	122°F		187°F		105.	

*Approximate.
Dieline (Stable isomers of Dichloroethylene)
Trieline (Trichloroethylene)
Carrene is a water white liquid at normal atmospheric conditions. It has a slightly sweetish odor similar to dieline, and is non-combustible.

The characteristics of the refrigerant are such that condenser water at 90° F. will produce condensation at an absolute pressure of about 9 lbs. This means that under extreme conditions, a pressure differential of approximately 7 lbs. exists between the intake and discharge side of the compressor. The ordinary pressure difference is usually less than 5 lbs.

From the condenser, the liquid refrigerant returns to the evaporator or cooler under its own head. This completes the very simple cycle. These simple operations are accomplished within a compact closed system and without the use of any valves in the refrigerant cycle.

The refrigerant used is "Carrene," a non-combustible, stable, water-white liquid at all normal atmospheric conditions, and may be handled freely in open containers as safely as water. The density of the vapor is 3½ times as great as that of air. The fact that the entire system operates under conditions of vacuum relative to the atmosphere, precludes possibilities of outward leaks; that is, the entire refrigeration cycle is completed under a vacuum. However, if any circumstances should cause the refrigerant to be liberated from the machine, its low diffusion characteristics are such that no hazards whatever would be offered to the public.

Referring to the diagrammatic sketch in Fig. 88—the complete cycle of refrigeration can be explained briefly as follows: The refrigerant (Dielene or Carrene, as the case may be) in a liquid state is contained in a reservoir at the bottom of the cooler (a). It is pumped by the liquid pump to a distributing pan directly over the tubes in the cooler (A). Here the liquid is allowed to shower down over the bronze tubes through which either brine (as in the Chicago Stadium for ice freezing) or water is circulated. This employs the very efficient flash-type cooling method. The difference in temperature between the medium within the tubes and the refrigerant causes it to vaporize and absorb latent heat at low temperature levels. The cooling effect is obtained by the free evaporation of the refrigerant liquid on the outside surfaces of the tubes. The vapor thus formed is drawn into the compressor where it is compressed to a higher absolute pressure, but still below atmospheric pressure, and discharged into the condenser chamber (B), where it is recondensed. Cold city water, river water, or water cooled by a cooling tower, passes through the condenser tubes, and the vapors surrounding are condensed to a normal liquid state and returned through a trap, hence back to the liquid chamber.

The characteristics of a centrifugal refrigeration system make it adaptable to automatic regulation of the cold water or brine temperatures regardless of the load. The general method of doing this is by controlling the flow of cooling water to the condenser. This is effected directly by thermostatic control. This means that the consumption of condenser water is always governed by the amount of refrigeration load, thus insuring the economical use of condenser water, and this is quite a factor where city water is being used. In the Chicago Stadium the maximum condenser water requirements are about 2 g.p.m per ton of refrigeration. Further regulation may be had by variable speed on the compressor, which reduces power required at

the reduced load. Such a requirement exists at Madison Square Garden, where during certain seasons, the machines are used to freeze ice, while at other times, the requirements are for cooling water to approximately 40° F. in conjunction with the air-conditioning system. It is claimed that the control is sufficiently accurate to permit the cooling of water to 35° F. without the danger of freezing. It is safe to guarantee a temperature fluctuation not to exceed 3° F. in the cooling chamber.

Reciprocating Pumps.—Pumps are used in refrigerating plants for pumping brine and water. The direct-acting pump is steam actuated and has no flywheel. This type of pump has few working parts, and is quite reliable and practical. It is built in the simplex and duplex types.

The simplex type has one steam cylinder and one brine or water cylinder. The steam valve, which is an ordinary slide valve, is actuated by various methods. In some types of pumps, the reversal of the stroke of the piston is obtained through linkage which is connected to the piston rods, while in the other cases the piston is reversed by means of small tappet valves which are placed at the ends of the cylinder. This type of pump, of course, is used in the smaller sizes only.

The duplex type of pump consists of two steam cylinders and two brine or water cylinders, which are placed side by side. The piston rod of one side controls the movement of the piston rod on the opposite side.

The brine and water ends may have different construction, such as the packed piston type, the center outside packed piston, and the outside packed pistons.

The center and outside packed plungers have the advantages of easy detection of leakage and ready adjustment of packing to suit the conditions.

Power pumps generally consist of three vertical single-acting water or brine plungers, which are driven by means of cranks and connecting rods. Power is usually supplied by an electric motor through a belt or silent chain, or it may be supplied by direct gearing.

Power pumps and direct-acting pumps have been used extensively in plants where brine or water has to be pumped considerable distance under fairly heavy pressure.

Steam pumps are quite uneconomical in the use of steam, using from 120 to 150 lbs. per hp. per hr.

The volumetric efficiency of reciprocating pumps is the ratio of the volume actually delivered to the plunger or piston displacement. Due to leakage past the valves and piston or plunger of the reciprocat-

ing pumps, the actual displacement is always greater than the quantity of brine or water delivered. The volumetric efficiency of reciprocating pumps in fair working condition may be assumed to be 85 to 90 per cent. The rating given in manufacturers' catalogues generally refers to the piston or plunger displacement, so that a deduction of 10 to 15 per cent must be made to cover the uusal slip and leakage. The mechanical efficiency of power pumps may be assumed to be 75 per cent, depending upon size of pump, refinement of design, etc.

Centrifugal Pumps.—In the centrifugal type of pump, the pressure is produced by the centrifugal force of water or brine which is set in motion in a closed casing by means of a bladed impeller. The centrifugal pump is particularly well adapted for direct connection to electric motors. It has few moving parts and no valves.

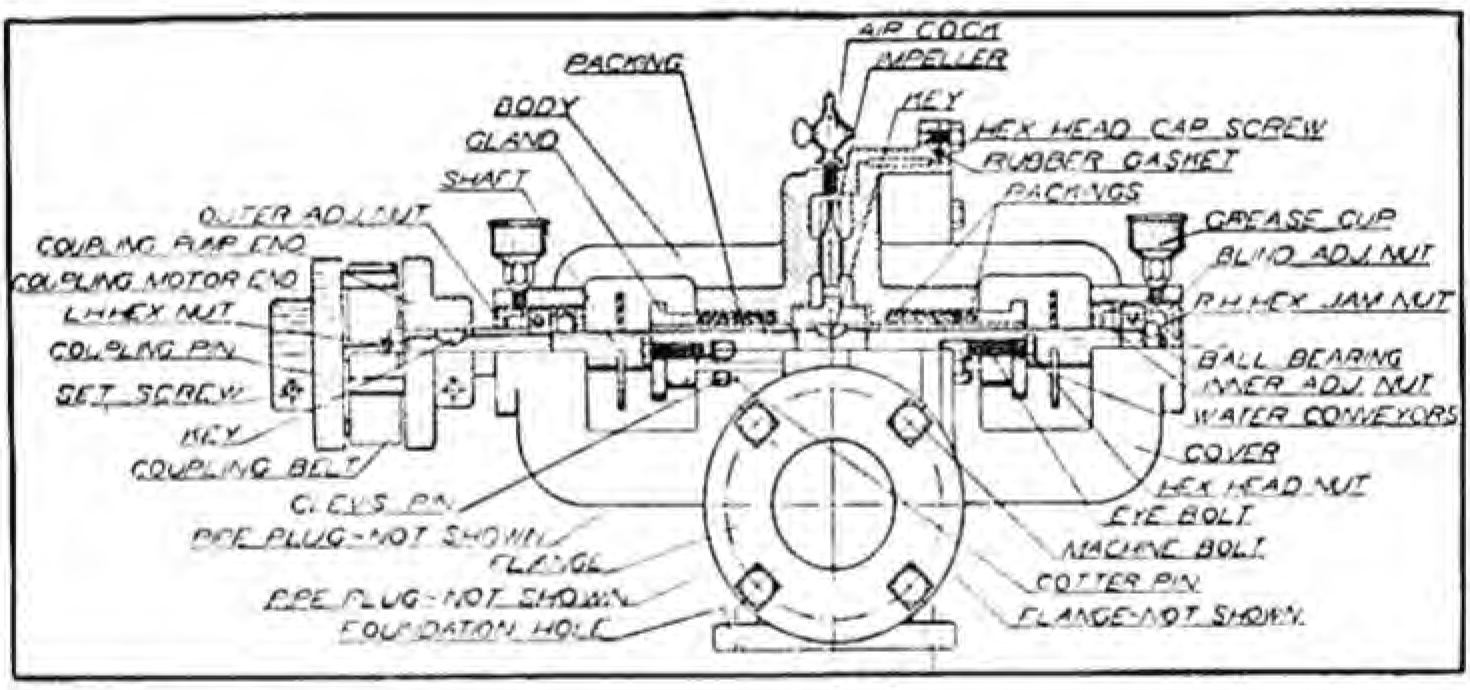


Fig. 91.—Sectional View of Westco Series of Single-Stage Pump.

However, in laying out or operating a centrifugal pump installation, it must be remembered that the efficiency of a pump depends upon such factors as the r.p.m., capacity and head. Centrifugal pumps are rated at their maximum efficiency at a given speed; thus, it is evident that these same conditions must be brought about in the plant in order to operate according to the manufacturers' specifications.

Figs. 91 and 92 show the construction used in the Westco turbine pumps.

In refrigeration plants, the centrifugal pumps have almost excluded the reciprocating pumps, except for very low capacities or high pressure. The mechanical efficiency is somewhat below that of the power pump and may be assumed to be 65 per cent, depending on the size of pump, the refinement of design, etc.

Power Required for Pumping.—The power required for pumping depends principally upon the weight of the liquid pumped and the vertical height to which it is raised from the source of supply to the point of delivery. It is also apparent that power is required to overcome the friction losses in the pump and the pipe lines. The power lost in the pump is determined from the mechanical efficiency, while power lost through friction in the lines must be estimated in some manner.

The work required to pump a fluid from one elevation to another is equal to the product of the weight in pounds and the height in feet. This product gives the number of foot pounds which are required, without allowance for losses. The horsepower may be determined by dividing the total number of foot pounds per minute by 33,000. This may be expressed in a formula as follows:

$$\text{Horsepower} = \frac{\text{Wt. of Liquid per Min. in Lbs.} \times \text{Height Pumped in Ft.}}{33{,}000}$$

$$\text{hp.} = \frac{W \times H}{33{,}000}$$

hp. = horsepower
W = weight of liquid pumped per min. in lbs.
H = total head or height in feet

The foregoing horsepower is the theoretical amount of power required for raising a fluid from one elevation to another. This must be increased in proportion to the mechanical efficiency of the pump in order to determine the brake horsepower of the pump; also, allowance must be made for the friction of the flow of the liquid in the pipe lines. The friction in the pipe lines may be taken to be the equivalent of additional elevation. In this case, the above formula may be written as follows:

$$\text{hp.} = \frac{W \times H}{33{,}000 \times E_m}$$

where H = total head in feet, including the loss due to friction
E_m = mechanical efficiency of the pump.

There are other losses in the pump, but these are only a small percentage of the total.

It is interesting, at this point, to observe how pressure may be changed into equivalent head or height of liquid. It is evident that a column of water will always produce a certain intensity of pressure upon its base. This intensity of pressure may be found by dividing the total pressure on the base by the area in square inches. Thus, if a column of water is 1 ft. high and has a base of 1 sq. ft., the pressure would be found by dividing the weight of the column, which is a cubic foot, by the area of the base, which is 144 sq. in.; or $62.5 \div 144 = 0.433$

lbs. pressure per sq. in. Thus, in any column of water of any height, each foot of head produces a pressure of 0.433 lbs. per sq. in.

In a similar manner, the pressure produced by brine may be determined. Since brine is heavier than water in direct proportion to the specific gracity, the pressure produced by each foot of head of brine would be equal to 0.433 × spec. grav. The above may be expressed in a formula as follows:

$$p = 0.433 \times h$$
$$p_1 = 0.433 \times h \times \text{spec. grav.}$$

where p = intensity of pressure due to water
p_1 = intensity of pressure due to brine or other liquid heavier than water
h = height or static head
spec. gravity. = specific gravity

Example 1.—It is desired to determine the brake horsepower of a pump to deliver 500 gal. of water per minute against the total head of

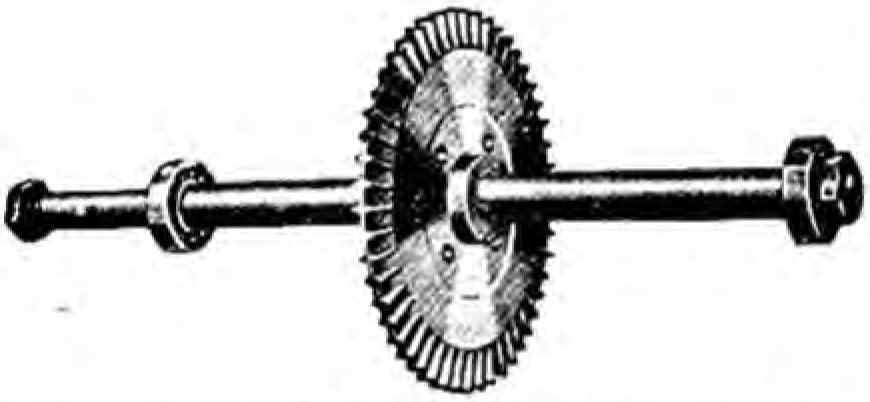


Fig. 92.—Hydraulic Balanced Multivane Impeller.

100 ft., when the mechanical efficiency of the centrifugal pump is 65 per cent. Here, the total water elevated per min. is equal to 500 × 8.33, since the weight of a gallon of water is 8.33 lbs. The horsepower is as follows:

$$\text{hp.} = \frac{(500 \times 8.33) \times 100}{33{,}000 \times 0.65}$$
$$\text{hp.} = 19.4$$

Example 2.—It is desired to determine the horsepower required for a centrifugal pump which delivers 200 gal. of brine per min. against a pressure of 50 lbs., when the specific gravity of the brine is 1.2 and the mechanical efficiency of the centrifugal pump is 65 per cent. The weight of brine per min., W, would be equal to 200 × 8.33 × 1.2 = 2,000 lbs. In this case, the head may be found as follows:

$$50 = 0.433 \times h \times 1.2$$
$$h = \frac{50}{0.433 \times 1.2} = 96.1$$

From this, the horsepower may be calculated as follows:

$$\text{hp.} = \frac{2{,}000 \times 96.1}{33{,}000 \times 0.65}$$

$$\text{hp.} = 8.95$$

Condenser Cooling Water Equipment.—One of the most important factors in the mechanical operation of refrigeration plants is the method of cooling the condenser water. The condenser water may

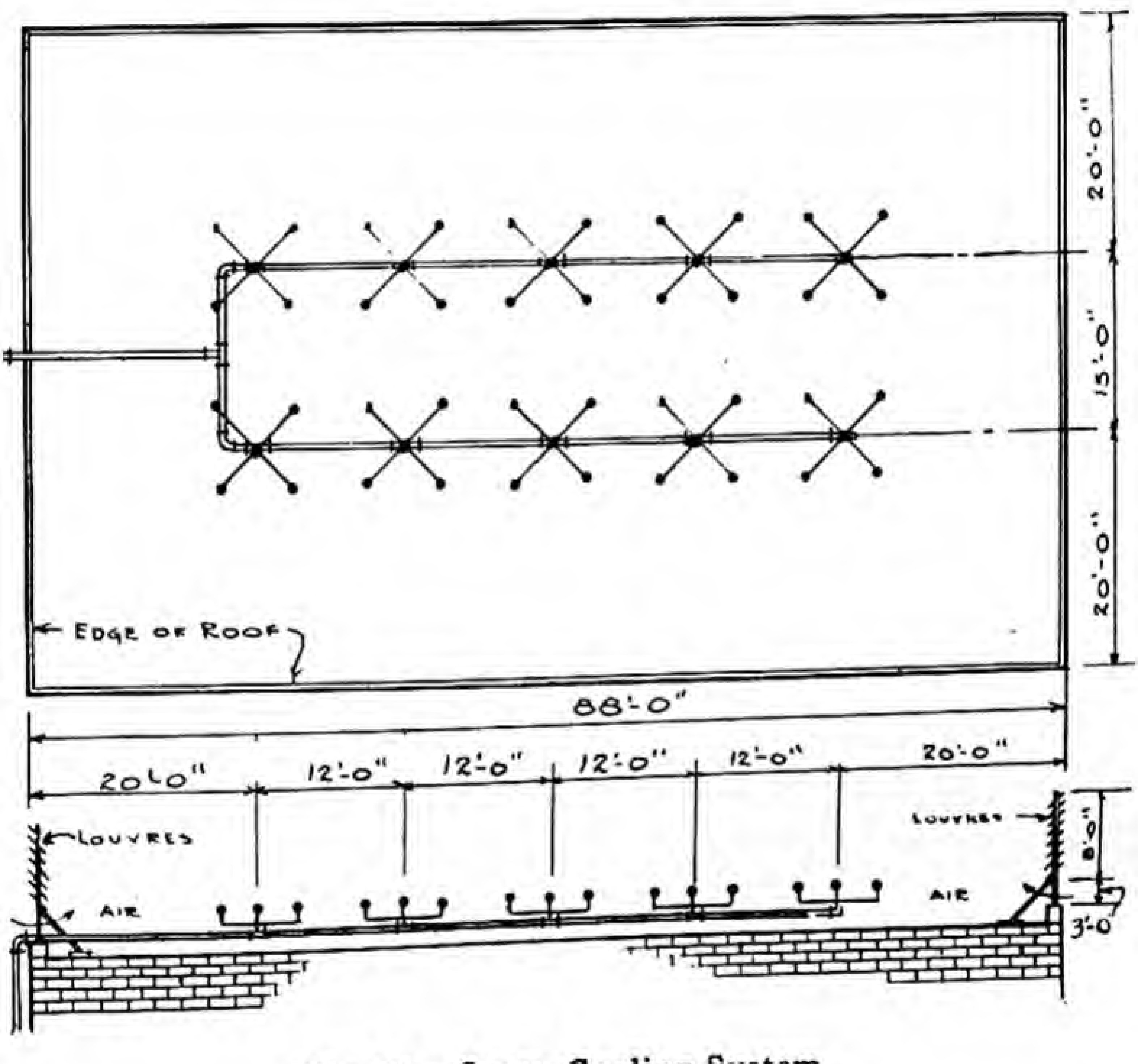


Fig. 93.—Spray Cooling System.

be taken from wells, lakes, rivers, and the sea. For economical reasons in the medium and large-sized plants, it is generally advisable to re-cool the condenser water, using it over and over again. This is due to the fact that the water supply for the ammonia condenser may be limited, or is obtainable only at a fairly high cost. It is evident, however, that each particular plant is an individual problem in itself.

There are several methods of re-cooling the condenser water, such as a large natural pond, a forced draft cooling tower, a natural draft

cooling tower, or a spray pond. In all of these methods of cooling the condenser water, the cooling is effected by heat flowing from the water to the air, thereby heating the air, and by evaporation of some of the water that is circulated.

In cooling towers and ponds, the water is divided into fine drops, and each of these drops is surrounded by moving currents of air. The air, in coming in contact with the fine drops of water, becomes heated and thereby cools the water, and also some of the water evaporates due to the absorption of heat, which thereby cools the total amount of the water a few degrees. On account of lack of space and water in the ordinary refrigeration plant, cooling ponds are seldom used. This method of cooling the water also has other disadvantages. At present cooling towers and spray ponds are used principally.

Spray Ponds.—One of the most efficient methods of cooling condenser water is that of spraying the heated water into the air above a suitable pond. This is accomplished by forcing the water through especially designed nozzles at a pressure of 5 to 10 lbs. for the purpose of breaking the water up into fine drops or spray. For comparatively short temperature ranges, the spray system provides a suitable means for cooling the condenser water. The small drops of water, upon falling through the air, become cooled, as previously indicated, so that the same water may be used over and over again. The amount of evaporation in general will amount to 1 to 2 per cent of the total amount circulated, and represents the amount that must be supplied from an outside source.

The general arrangement of a spray pond is shown by Fig. 93. This diagram shows the typical layout for a spray system which has fifty 1½-in. nozzles operating at a pressure of 7 lbs. gauge. The capacity of this system is approximately 1,000 gal. of water per minute.

Fig. 94, by the Cooling Tower Co., Inc., gives the general performance of spray cooling ponds when properly designed and installed.

The spray nozzles may be grouped upon roofs or above ponds on the ground. Generally, spray nozzles of the sizes known as 1½-in. and 2-in. are used. The nozzle openings in these sizes vary from ⅝-in. to ¾-in. These nozzles are generally arranged in groups of five, which are located 10 to 12 ft. apart. The water may be collected in a shallow pond of a depth of one to two feet, or may be collected upon a roof. When these ponds are installed on a roof, it is generally advisable to surround them with louvers for the purpose of preventing the wind from blowing too much of the water off the roof. The principal advantages of this method of cooling condenser water are the low initial cost and the low operating cost.

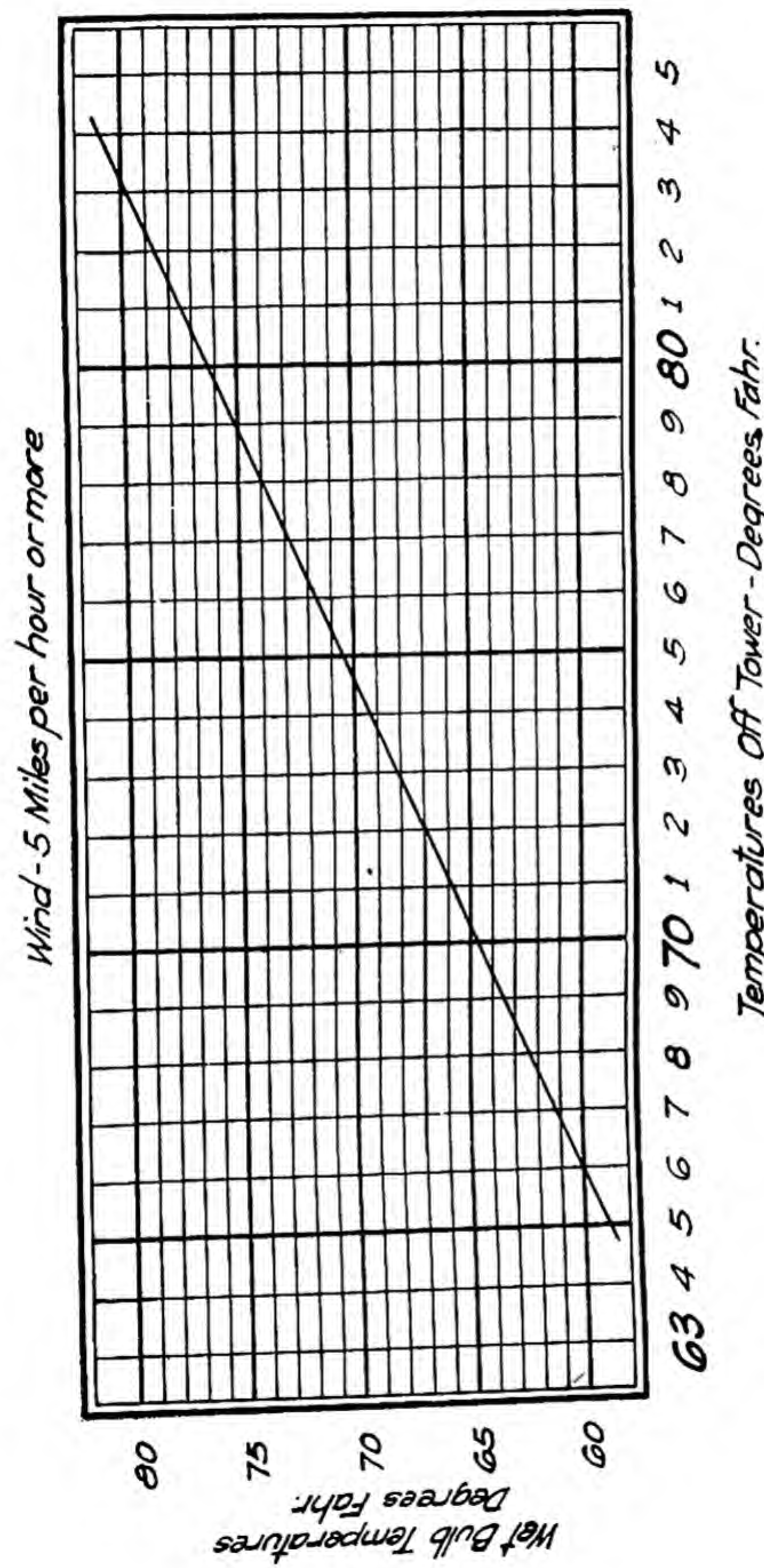


Fig. 94.—Cooling Curve for Condensing Service

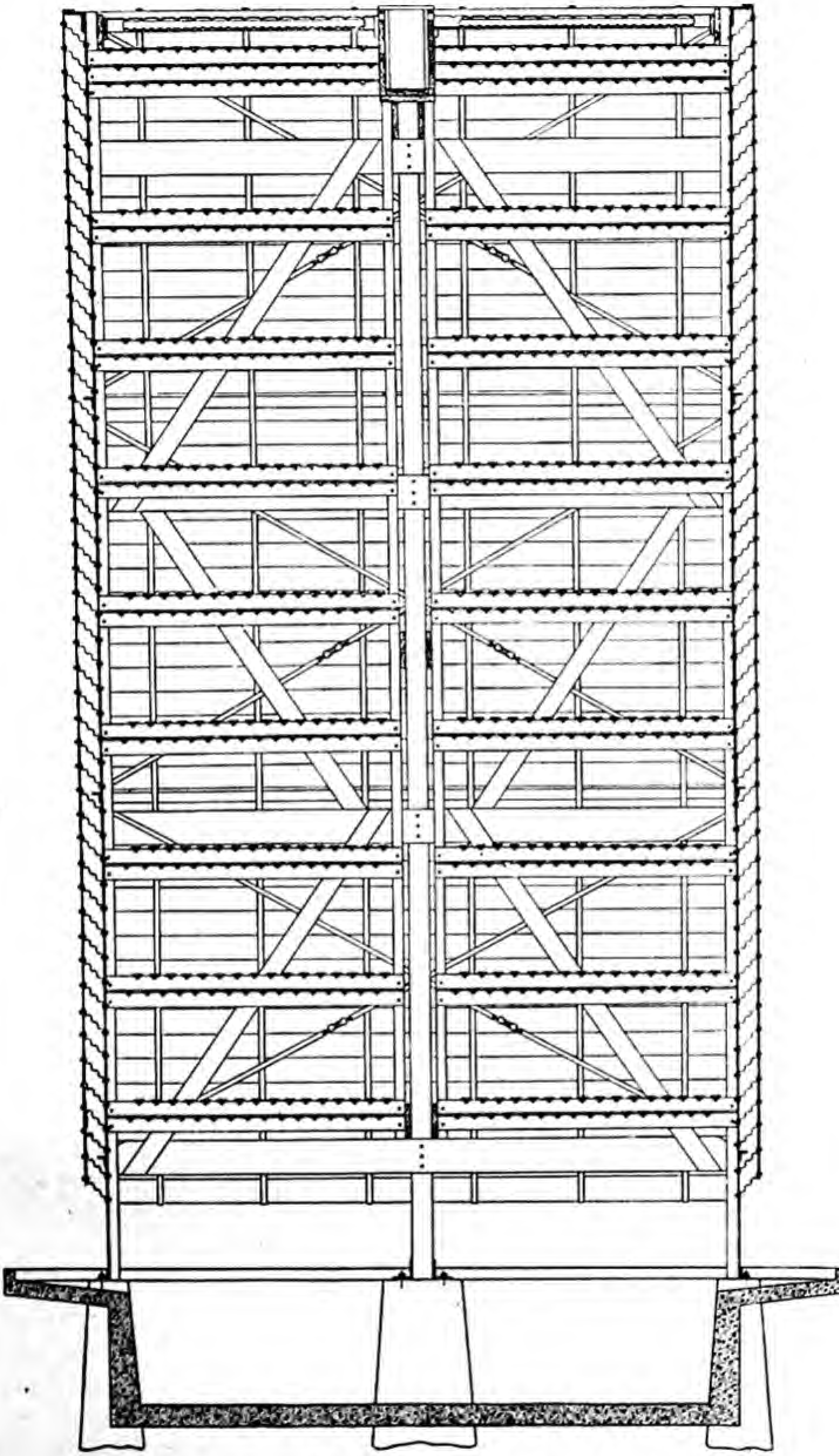


Fig. 95.—Cross-Section Showing Half Section of Cooling Tower Co. Towers.

The principal cost of the system is that of the piping and the nozzles, and since the operating pressures are quite low the horsepower required for pumping the water is not excessive. The spray system will cool the condenser water back to within 3 to 5 degrees of the wet bulb temperature. In general, the efficiency of the cooling will vary from 50 to 70 per cent. The efficiency, of course, depends upon the humidity of the air, the velocity of the air around the groups of spray nozzles, the fineness of the drops of water, etc.

Cooling Towers.—When there is not very much space available for cooling water equipment, the cooling tower may be used to an advantage. The cooling tower, as the name indicates, is nothing more than a tower which is generally filled with wooden or tile checker-work, or galvanized steel wire, etc., for the purpose of breaking the water up into fine drops and for retarding the falling of the water from the top of the tower to the bottom. The water, in passing down through the checker-work of the inside, presents a very large cooling surface to the rising currents of air. The water is cooled in the manner as previously indicated; that is, by heating the air and by evaporation of part of the water. The efficiency of the cooling will vary from 50 to 70 per cent as a general average. The water in general will be cooled 10° to 15° F., or will be cooled to within a few degrees of the wet bulb temperature.

The air is circulated through cooling towers either by means of the natural chimney action of the tower, or by the use of fans. The natural draft tower must necessarily be quite high. This is somewhat of a disadvantage. On the other hand, while the forced draft towers are comparatively low, power must be supplied for driving the fans. Furthermore, it is evident that energy must be expended in pumping the water to the top of the tower.

Fig. 95 illustrates type of tower made by Cooling Tower Co., Inc. The curves in Fig. 96 give the performance of the tower under various conditions.

Ordinarily, two sq. ft. of active horizontal deck area is recommended per ton of refrigeration.

Evaporative Condensers.—The installation of air conditioning systems in large city buildings brought on wider use of forced draft cooling towers because they require less space, have better appearance and lose less water in wind carry-out. The next step in logical development was to place the coils of a condenser inside the forced draft cooling tower and produce what is called an evaporative condenser. Essentially it consists of a metal housing with a compact coil of condenser pipe in the lower part, water sprays above and a blower

to circulate air through it. In operation the water is cooled on the surface of the condenser at the same time the refrigerant is being condensed. In the winter time a number of these condensers are operated on air circulation alone and there is no danger of freezing up the water portion of the apparatus.

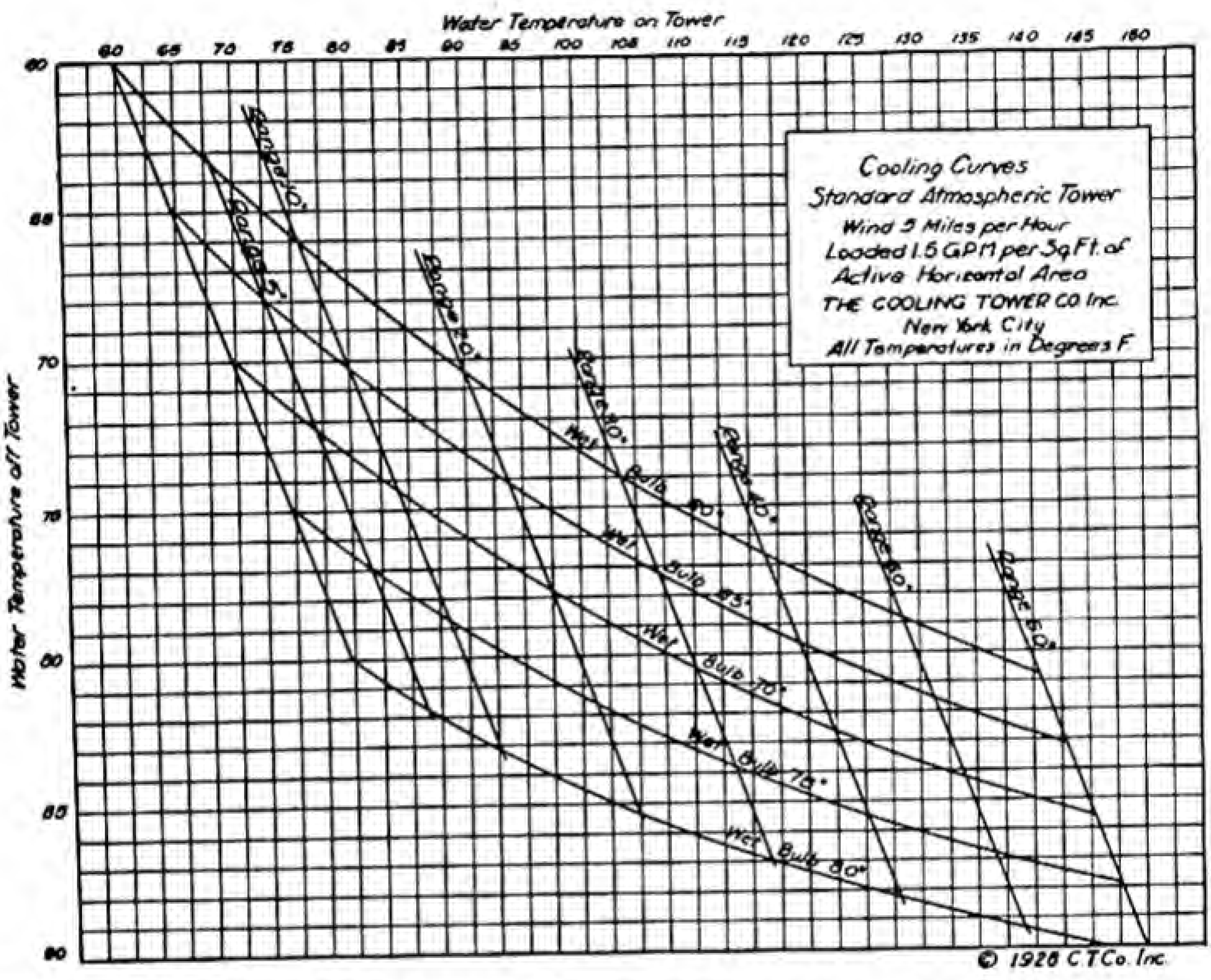


Fig. 96.—Chart Showing Performance of Tower Under Various Conditions.

If desired, evaporative condensers can be placed inside buildings with the air discharge arranged to be carried outside. The water evaporated in the cooling process is replenished through a float valve in the sump at the bottom of the evaporative condenser and the overall results obtained with the equipment have brought about many installations, even where space was available for other types of cooling towers and condensers. Scale problems can pose difficulties with evaporative condensers and the character of the cooling water should be considered along with proposed installations of them.

While the evaporative condenser requires a blower to circulate the air through it the motor on the water pump is much smaller than with a cooling tower. A cooling tower and conventional condenser will have about the same installed cost as an evaporative condenser of equal capacity.

QUESTIONS ON CHAPTER VII

1. Describe briefly the three principal types of absorption refrigerating machines.

2. What factors determine the type of absorption machine to be used in any particular plant?

3. What are the principal differences between construction of the ammonia and the carbon dioxide compression systems?

4. Describe the principle of operation of the "Electrolux" refrigerating machine.

5. Describe the use of solid carbon dioxide as a refrigerant.

6. Give a brief description of the Carrier "Centrifugal" refrigerating system.

7. Why is the centrifugal pump used more than the reciprocating pump for pumping brine and water?

8. Describe three methods of re-cooling refrigerant condenser water.

9. When the atmospheric temperature is 90° F. (dry-bulb), how is a cooling tower or spray pond able to cool water to several degrees lower than 90°?

10. What are some of the advantages of the evaporative condenser?

CHAPTER VIII.

HEAT TRANSMISSION IN INSULATION AND APPARATUS.

General Consideration.—In order to retard or lessen the flow of heat from the exterior to the interior of refrigerated rooms, spaces, or substances, a material having pronounced resistance to the flow of heat is interposed between the regions of the different temperatures. Such materials are termed "non-conductors of heat," since heat passes through them at a very slow rate. Thus, since it is obviously impossible to construct a wall so as to entirely prevent the inflow of heat, it is evident that the heat will flow continually from the warmer exterior regions through the insulated wall into the cooler interior regions.

Therefore, suitable means must be provided for producing refrigeration in the room or space of low temperature, to offset or remove the heat which is transmitted through the insulated wall. Furthermore, it is evident that the use of an insulated wall will retard the outflow of heat from cooled rooms and spaces to the colder exterior during the periods of extremely low temperatures in the winter season.

Broadly speaking, the use of insulating materials is an economic consideration. There are great economic advantages of reducing to a minimum the loss of refrigeration through insulated walls; these are the reduction of the operating cost and lowering of the initial cost of the refrigeration plant. If the walls are poorly insulated, the refrigeration required to offset the heat transmitted through the insulation is large; this increases the daily operating cost. Likewise, a larger refrigerating machine must be used; thus the initial cost of the refrigeration plant is greater.

The problem of efficiently securing and maintaining low temperatures is a proposition of constructing a wall that will resist or retard the rapid inflow of heat. The importance of constructing a wall that is a non-conductor of heat will be more fully understood when it is realized that about three-quarters of the refrigeration load on a cold storage refrigerating plant is made up of heat which is transmitted through the insulation.

In order to reduce the transmission of heat to a minimum, thick walls of ordinary materials may be used, or thin walls with efficient insulation may be used.

Heat Transmission.—Heat is transmitted from a region of higher temperature to a region of lower temperature by the natural and continual tendency of heat toward temperature equilibrium. When temperature equilibrium exists, or when there is no temperature difference, there can be no transfer of heat. When there is a temperature difference, the flow of heat is always toward the lower temperature level.

The rate of heat transfer from one region to another depends upon the area of the heat transmitting surface, the mean temperature difference, and the unit heat transfer coefficient. This heat transfer coefficient is the unit overall heat transfer rate, generally expressed in Btu. per hour per degree of mean temperature difference. The magnitude of the heat transfer coefficient depends upon the heat that may be transmitted by radiation, convection, and conduction.

The heat transfer coefficient, therefore, combines the three modes of heat transfer into a single quantity. The actual magnitude of the heat transfer coefficient is determined in practice by calculation based upon theoretical analysis and experimentation.

The fundamental heat transfer law may be expressed in symbols as follows:

$$H = K \times A \times t_d$$

where H = total heat transfer in Btu. per hour
K = heat transfer coefficient, Btu per hour. per deg. temp. diff.
A = area of heat transmitting surfaces, in sq. ft.
t_d = mean temp. diff.

In other words, the actual amount of heat transmitted in any case depends upon the product of the coefficient, area, and temperature difference. In the case of a cold storage room wall, where the temperatures are constant, the heat transfer may be found as follows:

$$H = K \times A \times (t - t_o)$$

where t = temperature of outside warm air
t_o = temperature of inside cold air.

From the foregoing it will be noted that, if the temperature and area of heat transmitting surface are known and are held constant, the heat transfer depends upon radiation, convection, and conduction.

Conduction.—Heat transfer by conduction is accomplished by means of molecular communication. Heat is carried through a material by means of the vibrations of the molecules themselves. Fig 97 shows graphically how this may be done. This figure shows a section of a material, the molecules of the material being represented by the

small circles. The molecule A at the warm surface vibrates in proportion to its temperature of 70° F. The molecule A strikes the molecule B, which in turn strikes molecule C, and so on, until the vibrations are transmitted through the insulation to the molecule E, which has a vibration corresponding to a heat intensity of 30° F.

Due to molecular friction, adhesion, etc., it is evident that the vibration will become slower as the energy passes from one molecule to the other, so that the temperatures will be lower, as indicated by Fig. 97. Thus, heat may be carried through a metal pipe, a brick wall, wood, paper, or cork. An iron rod, held in a fire, will become heated at the opposite end due to conduction. In general, metals are good conductors of heat, while lighter materials are poor conductors. Thus, the efficiency of insulating materials may be roughly judged by noting their densities.

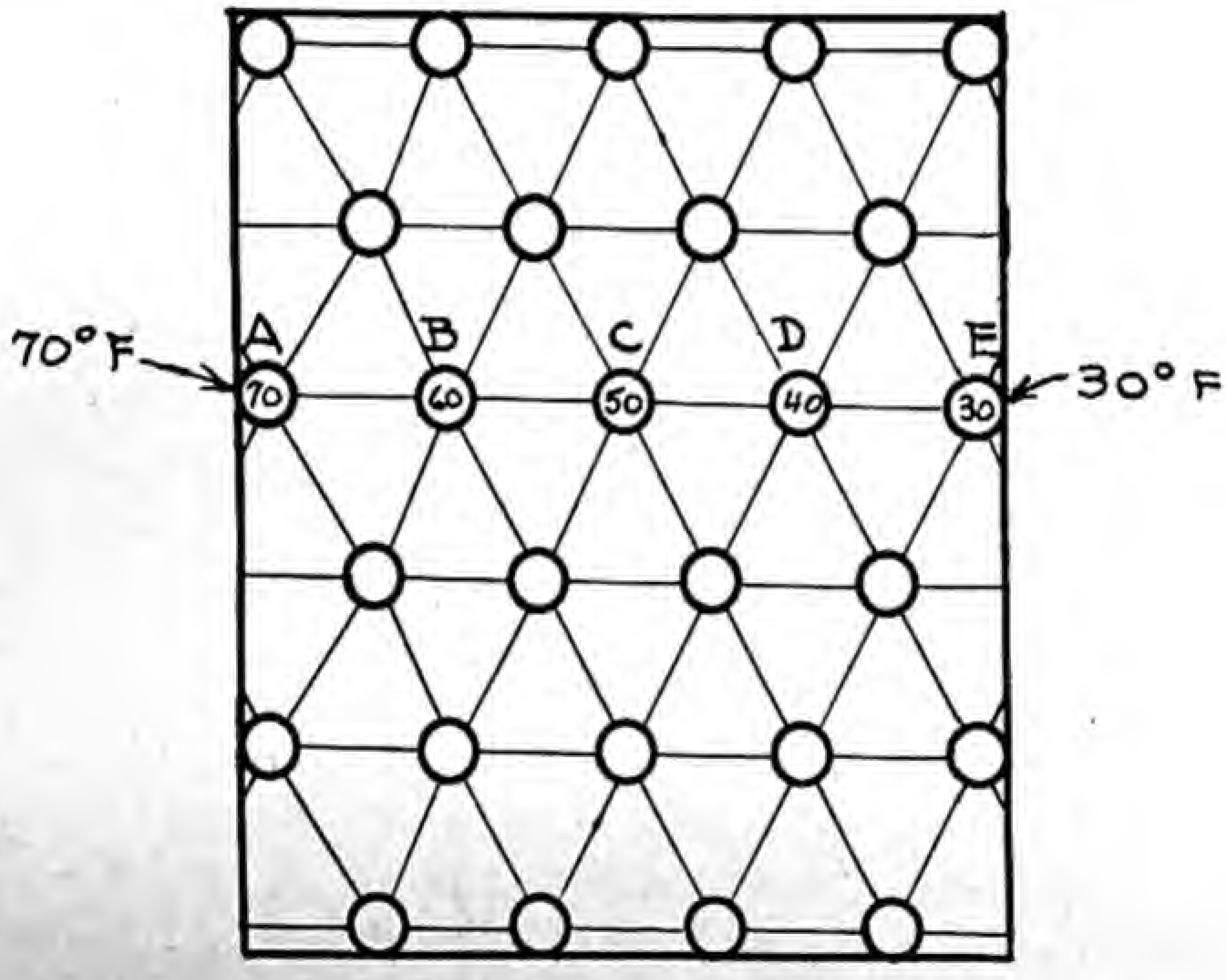


Fig. 97.—Conduction of Heat.

The amount of heat that will be transmitted through a given material due to a given temperature difference depends upon the characteristic internal thermal conductivity of the material.

The conductivity of a material is then the rate of heat transmission through the material by the molecular vibration or, in short, by conduction, and every material has its own characteristic rate of conduction. The amount of heat that will be transmitted through a material having parallel surfaces depends upon the temperature difference, kind, and thickness of material. The condition and relative

temperature of the material also affect the rate of conduction. In general, the heat transmitted by conduction may be found as follows:

$$H_1 = \frac{C}{X}(t_1 - t_2)$$

where H = heat transmitted by conduction
C = coefficient of internal conductivity in Btu. per sq. ft. per hr. per deg. temp. diffi. per in. of thickness of material
t_1 = high (surface) temperature
t_2 = low (surface) temperature
X = thickness of material in inches

The internal thermal conductivities of various materials will vary with their conditions, densities, etc. The internal conductivities of various materials under testing laboratory conditions are given in Table 57. These are based upon the results of experiments of the United States Bureau of Standards and others. Column 3 gives the internal conductivities in Btu. per twenty-four hours, while column 4 gives the conductivity in Btu. per hour. The approximate weight per cu. ft. is given by column 5.

The heat transmitted through a 4-in. corkboard, having surface temperatures of 70° F. and 30° F., may be considered. Here, $t_1 = 70$, $t_2 = 30$, $X = 4$, and C (from Table 57) = 0.308. Thus, the heat transmitted by conduction is:

$$H_1 = \frac{0.308}{4}(70 - 30) = 3.08 \text{ Btu. per hour.}$$

Radiation.—Radiation is the transfer of heat due to the entire or partial transformation of the energy of light into the heat energy by impact upon the surface of a substance. It is in this way that heat is transmitted from the sun to the earth, from burning fuel or red-hot stoves to other objects. The amount of heat transferred by radiation depends upon the characters of the hot and cold surfaces, the temperature differences, the relative absolute temperatures, the absorbing power of the cold surfaces, etc.

Convection—Convection is the transfer of heat by displacement of movable media. Thus, heat may be transmitted in this manner by liquids and gases, since the heated medium moves and thus carries heat energy with it. The natural circulation of air in a cold storage room is an example of this mode of heat transfer. The heat transferred by this means depends upon the kind of fluids, temperature differences, velocity of fluids, character of surfaces, etc.

Radiation and Convection.—From the foregoing, it is evident that many variables enter into the determination of heat transfer by radiation and convection. Therefore, reliable experimental information

TABLE 57.—INTERNAL THERMAL CONDUCTIVITIES OF VARIOUS MATERIALS, PER SQ. FT. PER IN. THICK. PER DEG. T.D.

Material	Description	Btu. per 24 hours	Btu. per hour	Lb. per cu. ft.
Air	Ideal air space	4.2	0.175	0.08
Air Cell, ½ inch	Asbestos paper and air spaces	11.0	0.458	8.80
Air Cell, 1 inch	Asbestos paper and air spaces	12.0	0.500	8.80
Aluminum	Cast	24.000	1000.000	168.50
Ammonia Vapor	32° F	3.19	0.133	0.21
Aqua Ammonia	64° F	75.90	3.160	56.50
Asbestos Mill Bd.	Pressed asbestos—not very flexible	20.00	0.830	61.00
Asbestos Paper	Asbestos and organic binder	12.	0.500	31.0
Asbestos Wood	Asbestos and cement	65.0	3.700	123.0
Balsa Wood	Very light and soft—across grain	8.4	0.350	7.5
Boiler Scale		305	12.700	.....
Brass		15,000	625.000	250.
Brick	Heavy	120	5.000	131.
Brick	Light, dry	84	3.500	115.
Brine	Salt	27.1	1.130	73.4
Cabot's Quilt	Eel grass enclosed in burlap	7.7	0.321	16.0
Calorax	Fluffy finely divided mineral matter	5.3	0.221	4.0
Celite	Infusorial earth powder	7.4	0.308	10.6
Cement	Neat Portland, dry	150.0	6.250	170.
Charcoal	Powdered	10.0	0.417	11.8
Charcoal	Flakes	14.6	0.613	15.0
Cinders	Anthracite, dry	20.3	0.845	40.0
Concrete		125.0	5.200	136.0
Concrete	Of fine gravel	109.0	4.540	124.0
Concrete	Of slag	50.0	2.080	94.5
Concrete	Of granulated cork	43.	1.790	7.5
Copper		50,000	2083.000	556.0
Cork	Granulated ⅛-3/16 inch	8.1	0.337	5.3
Cork	Regranulate 1/16-⅛ inch	8.0	0.333	10.0
Corkboard	No artificial binder—low density	6.7	0.279	6.9
Corkboard	No artificial binder—high density	7.4	0.308	11.3
Cotton Wool	Loosely packed	7.0	0.292	.....
Cypress	Across grain	16.0	0.666	29.0
Fiberglas		5.76	0.24	3.0
Fire Felt Roll	Asbestos sheet coated with cement	15.0	0.625	43.0
Fire Felt Sheet	Soft, flexible asbestos sheet	14.0	0.583	26.0
Flaxlinum	Felted vegetable fibers	7.9	0.329	11.3
Fullers Earth	Argillaceous powder	17.0	0.708	33.0
Glass		124.0	5.160	150.0
Glass		178.0	7.420	185.0
Granite		600	25.000	166.0
Granulated Cork	About 3/16 inch	7.5	0.313	8.1
Gravel	Dry, coarse	62.0	2.582	115.0
Gravel	Dry, fine	39.0	1.630	91.25
Ground Cork		7.1	0.294	9.4
Gypsum Plaster		54.0	2.250	.....
Hair Felt		5.9	0.246	17.0
Hard Maple	Across grain	27.0	1.125	44.0
Ice		408	17.000	57.4
Infusorial Earth	Natural blocks	14.0	0.583	43.0
Insulex	Asbestos and plaster blocks—porous	22.0	0.916	29.0
Insulite	Pressed wool pulp—rigid	7.1	0.296	11.9
Iron	Cast	7,740	321.500	450.0
Iron	Wrought	11,600	483.000	485.0
Kapok	Imp. vegetable fiber — loosely packed	5.7	0.238	0.88
Keystone Hair	Hair felt confined with building paper	6.5	0.271	19.0
Limestone	Close grain	368	15.300	185.0
Limestone	Hard	214.0	9.330	159.0

TABLE 57.—INTERNAL THERMAL CONDUCTIVITIES OF VARIOUS MATERIALS PER SQ. FT. PER IN. THICK PER DEG. T.D. —(Concluded.)

Material	Description	Btu. per 24 hours	Btu. per hour	Lb. per cu. ft.
Limestone	Soft	100.0	4.167	113.0
Linofelt	Vegetable fiber confined with paper	7.2	0.300	11.3
Lithboard	Mineral wool and vegetable fibers	9.1	0.379	12.5
Mahogany	Across grain	22.0	0.916	34.0
Marble	Hard	445	18.530	175.0
Marble	Soft	104	4.330	156.0
Mineral Wool	Medium Packed	6.6	0.275	12.5
Mineral Wool	Felted in blocks	6.9	0.288	18.0
Oak	Across grain	24.0	1.000	38.0
Paraffin	"Parowax," melting point 52° C.	38.0	1.582	56.0
Petroleum	55° F.	24.7	1.030	50.0
Plaster		132.0	5.500	105.0
Plaster	Ordinary mixed	90	3.750	83.5
Plaster	Board	73	3.040	75.0
Planer Shavings	Various	10.0	0.417	8.8
Pulp Board	Stiff pasteboard	11.0	0.458	......
Pumice	Powdered	11.6	0.483	20.0
Pure Wool		5.9	0.246	6.9
Pure Wool		5.9	0.246	6.3
Pure Wool		6.3	0.263	5.0
Pure Wool		7.0	0.292	2.5
Redwood Bark		6.13	0.255	4.0
Rock Cork	Mineral wool and binder—rigid	8.3	0.346	21.0
Rubber	Soft	45	7.875	94.0
Rubber	Hard, vulc.	16.0	0.667	59.0
Sand	River, fine, normal	188.0	7.830	102.0
Sand	Dried by heating	54.0	2.250	95.0
Sandstone		265	11.100	138.0
Sawdust	Dry	12.0	0.500	13.4
Sawdust	Ordinary	25.0	1.040	16.0
Shavings	Ordinary	17.0	0.707	8.0
Silicate Cotton		14.0	0.583	8.55
Slag Wool		18.0	0.750	15.0
Snow on Ref. Coils		75	3.130	.....
Tar Roofing		17.0	0.707	55.0
Vacuum	Silvered vacuum jacket	0.1	0.004	.....
Virginia Pine	Across grain	23.0	0.958	34.0
Water	Still, 32° F.	100	4.166	62.4
White Pine	Across grain	19.0	0.791	32.0
Wool Felt	Flexible paper stock	8.7	0.363	21.0

regarding these is lacking, since it is difficult to ascertain the exact effect of each. Also, in the ultimate analysis, it will be noted that the refrigerating engineer is concerned, not with the proportion of heat transferred by conduction, radiation, or convection, but the total overall heat transfer.

The combined transfer of heat by radiation and convection may be determined by experimentation. The combined coefficient or rate of this heat transfer is the heat given off or absorbed per square foot of surface per hour per degree of temperature difference. In the case of cold storage insulation, the temperature difference would be taken as the difference between the temperature of the surface of the wall and the average air temperature. Likewise, it is evident that the velocity of the air across the surface of the insulation will affect the magnitude of the coefficient of radiation and convection.

The values for the coefficient of radiation and convection for various insulating materials under still air conditions are shown by Table 58. These coefficients are the heat transfer in Btu. per hour per degree of temperature difference. These values are based upon experiments made at the Engineering Experiment Station of the University of Illinois.

This coefficient is generally denoted by the symbol K_1, and is called the coefficient of radiation and convection for inside surfaces. In an actual plant, the outside walls are exposed to the more rapid movement of the air, so that the coefficient of radiation and convection is larger for the outside surfaces. The symbol for this coefficient is K_2, and it is generally 2.5 to 3 times the inside wall coefficient K_1, due to the greater velocity of the air. Thus, as a general rule, the value of outside coefficient K_2, may be considered to be three times the inside coefficient, K_1.

TABLE 58.—COEFFICIENTS OF RADIATION AND CONVECTION (K_1) IN BTU. PER HOUR PER DEGREE TEMPERATURE DIFFERENCE.

Material	Coefficient K_1
Brick wall	1.40
Concrete	1.30
Wood	1.40
Corkboard	1.25
Magnesia board	1.45
Glass	2.00
Tile plastered on both sides	1.10
Asbestos board	1.60
Sheet asbes.os	1.40
Roofing	1.25

Total Heat Transfer Through an Insulated Wall.—From the foregoing, it will be observed that heat may be transmitted from a region of high temperature through an insulated wall into a region of lower temperature by means of radiation, convection, and conduction. Fig. 98 shows graphically the transfer of heat through a wall. Here, the heat passes by radiation and convection from a warm surface at t° F. to the outside surface of the wall. It is absorbed by the outside surface, conducted through the material, and given off by the inside surface by means of radiation and convection to the cold surface at t_0° F.

As previously indicated, the heat will be conducted or transmitted through the wall, due to the temperature difference. In Fig. 98, this is the difference between the temperature at the surface on the warm side, t_1, and the temperature at the surface on the cold side, t_2. The amount of heat transferred, as previously indicated, would depend upon the internal thermal conductivity and the thickness of the material.

In a similar manner, since the heat that is conducted through the wall or material must be equal to the amount absorbed by the warm surface and that given off by the cold surface, it is evident that there must be temperature differences at the surfaces of the material in order to cause the heat transfer. This temperature difference exists within very thin layers of air at the surfaces.

On the warm side of the wall in Fig. 98, this is represented by the differences between the temperature of the outside air, t, and the temperature at the outside surface, t_1, and on the cold side the difference is between the temperature of the inside surface, t_2, and the temperature of the inside air, t_0.

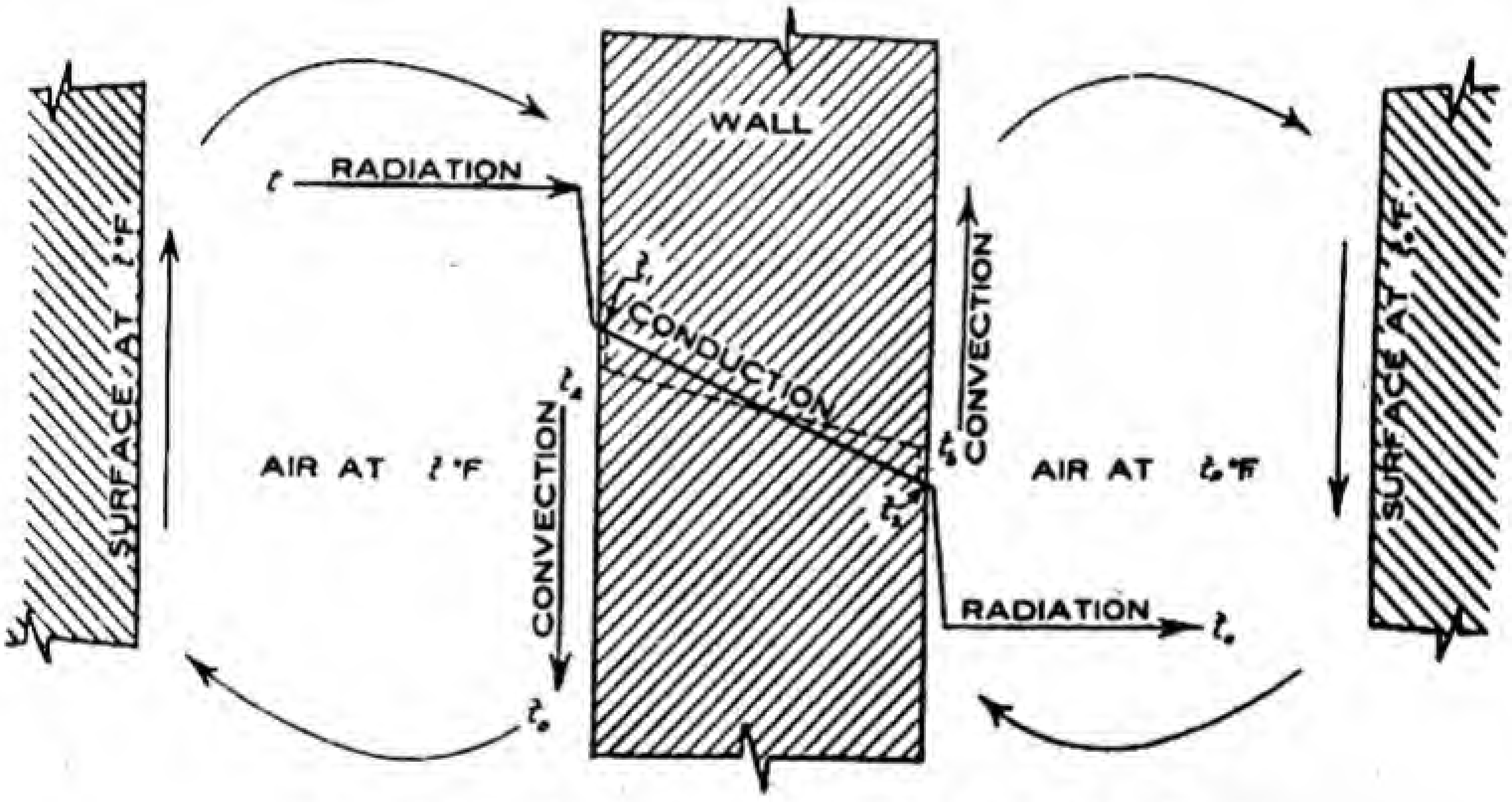


Fig. 98.—Heat Transfer Through a Wall.

The amount of heat transferred depends upon the radiation and convection effects. It is evident that the quantity of heat transferred from the warm air to the wall will depend upon the coefficient of radiation and convection, K_2. This is sometimes termed the surface coefficient.

In a similar manner, the heat given off by the inside surface to the air will be observed to depend upon the coefficient of radiation and convection for the inside surface, K_1, and the temperature of the inside surface, t_2, and the temperature of the air, t_0.

The total heat transmission through the wall will depend upon the overall heat transfer coefficient, K, and the temperature of the warm air, t, and the temperature of the cold air, t_0. From the foregoing analysis, the value of the unit overall heat transfer coefficient may be determined and is as follows:

$$K = \frac{1}{\frac{1}{K_1} + \frac{X}{C} + \frac{1}{K_2}}$$

From this formula, it will be observed that the overall heat transfer coefficient, K, in Btu. per hour per degree temperature difference for a given wall depends upon the radiation and convection coefficients, K_1 and K_2, the thickness of the wall, X, and the internal conductivity of the material, C. The values of the conductivity, C, for various materials are given by Table 57, while the values of the coefficients of radiation and convection, K_1, may be found in Table 58. The values of K_2 may be taken as three times K_1.

In general, the value of the quantity $\frac{X}{C}$ for thin walls composed of good conductors of heat becomes so small that it may be neglected, while in the case of a thick wall of good insulation, the value of $\frac{1}{K_1}$ and $\frac{1}{K_2}$ become so small that they may be neglected.

In case of a solid compound wall with materials having different conductivities, C_1, C_2, C_3, etc., of various thicknesses, X_1, X_2, X_3, etc., the above formula is written as follows:

$$K = \frac{1}{\frac{1}{K_1} + \left\{\frac{X_1}{C_1} + \frac{X_2}{C_2} + \frac{X_3}{C_3} + \text{etc.}\right\} + \frac{1}{K_2}}$$

In the event that the compound wall contains air spaces, coefficients for still air condition for each material and surface must be used, excepting that the coefficient of the outside wall must be increased. The above formula, then, is written as follows:

$$K = \frac{1}{\frac{1}{K_1} + \left\{\frac{X_1}{C_1} + \frac{X_2}{C_2} + \frac{X_3}{C_3} + \text{etc.}\right\} + \frac{1}{K_2} + \frac{1}{K_3} + \text{etc.}}$$

Example 1. It is desired to determine how much heat is transmitted through an outside brick wall 13 in. thick, 10 ft. high, and 50 ft. long, when the outside temperature is 80° F. and the inside temperature is 30° F. per hour, per Fig. 99.

From Table 57 it will be found that, $C = 5$; from Table 58, $K_1 = 1.40$, and since $K_2 = 3 \times K_1$, $3 \times 1.40 = 4.20$. Thus, the heat transmission coefficient, K, is found as follows:

$$K = \frac{1}{\frac{1}{1.4} + \frac{13}{5} + \frac{1}{4.2}} = 0.282 \text{ Btu. per hr. per deg. temp. diff.}$$

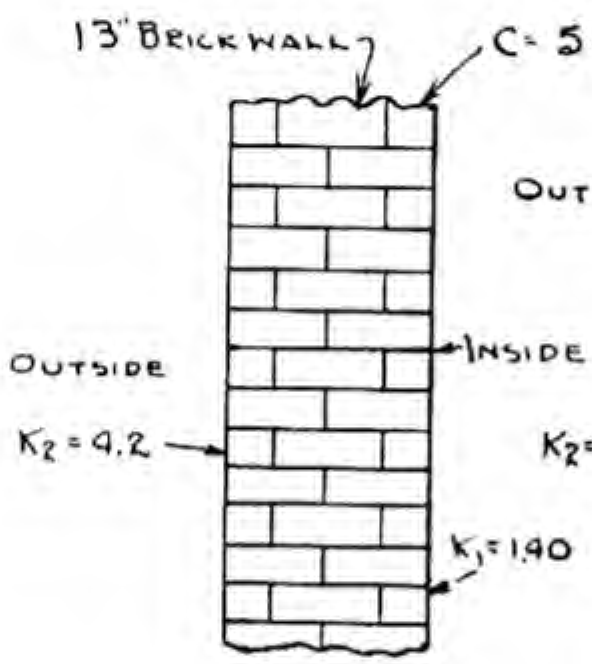


Fig. 99.

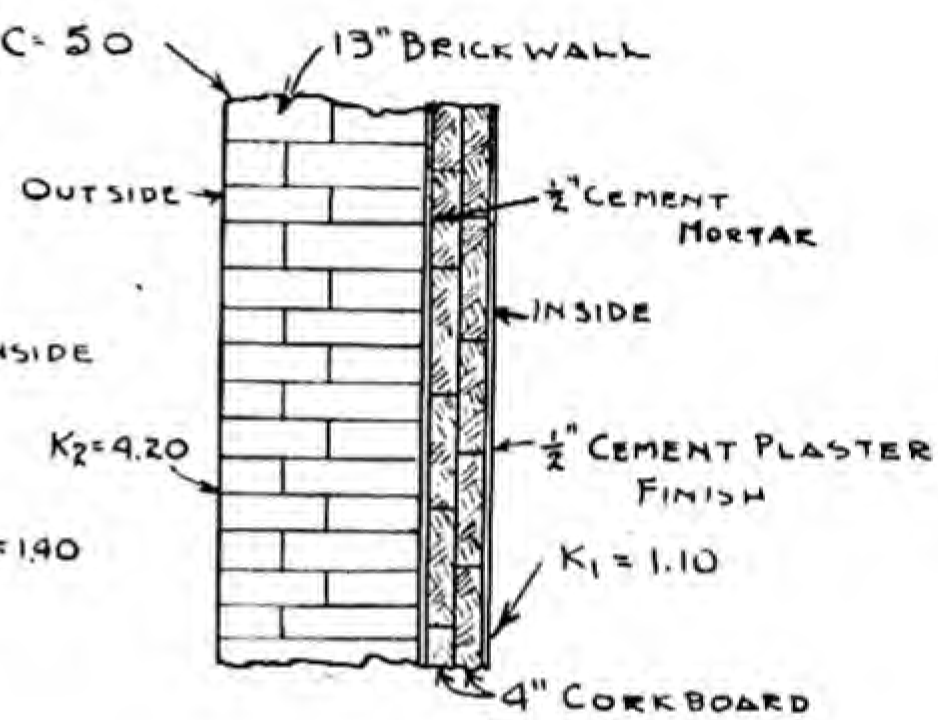


Fig. 100.

The area, A, is equal to $50 \times 10 = 500$ sq. ft., and $t_1 = 70°$ and $t_2 = 30°$. The total heat transfer is found as follows:

$$\begin{aligned} H &= K \times A \times (t_1 - t_2) \\ &= 0.282 \times 500 \times (70 - 30) \\ &= 5640 \text{ Btu. per hr.} \end{aligned}$$

This is equivalent to $5640 \div 12000 = 0.47$ tons of refrigeration per day.

Example 2. It is desired to determine the heat transmission of the brick wall in the foregoing problem, if it is insulated with four inches of corkboard applied to the wall and finished with coatings of cement mortar one-half inch thick, per Fig. 100. The value of K is found as follows:

$$\begin{aligned} K &= \frac{1}{\frac{1}{1.10} + \frac{1}{4.20} + \frac{13}{5} + \frac{4}{0.308} + \frac{1}{6.25}} \\ &= 0.0592 \text{ Btu.} \end{aligned}$$

Then H is found as follows: $H = 0.0592 \times 500 \times (70° - 30°)$
$= 1184$ Btu. per hr.

This is equivalent to 1184 ÷ 12,000 = 0.0985 tons of refrigeration per day.

Example 3. It is desired to determine the heat transfer coefficient, K, of a 10-in. concrete wall, insulated with eight 1-in. boards and air spaces as indicated by Fig. 101. The value of K is found as follows:

$$K = \frac{1}{8\left\{\frac{1}{1.40}\right\} + \frac{10}{5.21} + \frac{8}{1} + \frac{1}{3.90} + \frac{1}{1.30}}$$
$$= 0.060 \text{ Btu.}$$

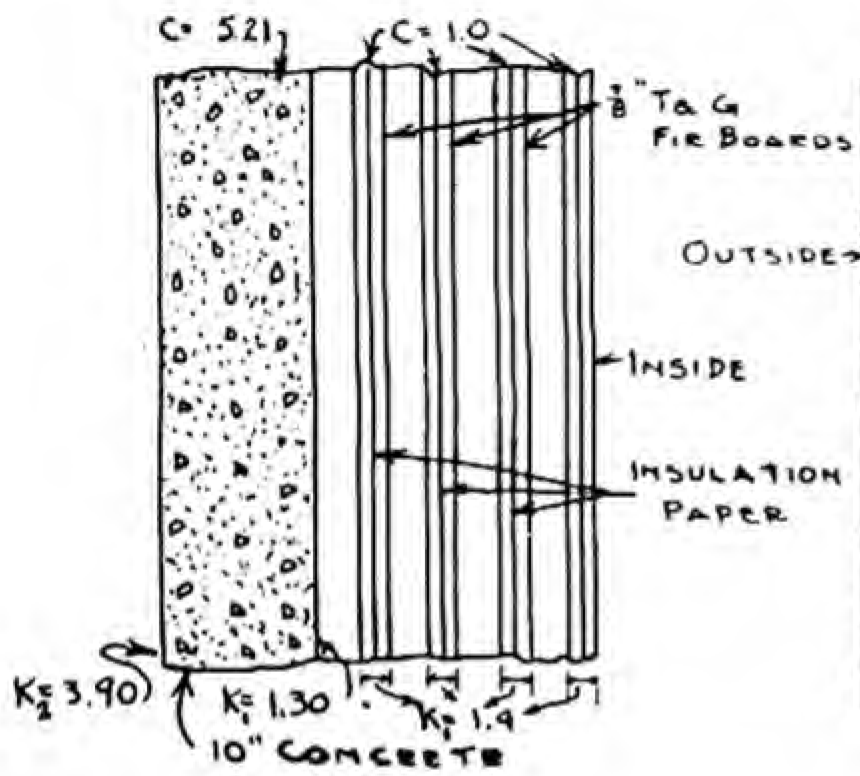


Fig. 101.

Fig. 102.

It should be observed that the determination of the amount of heat transmitted through an insulation depends upon its particualr application in the plant and that the estimate of the heat transfer must be based upon the actual working conditions. In order to obtain an estimate of the amount of heat that could possibly flow through the wall, the amounts as calculated in the foregoing should be increased from 25 to 35 per cent. This is due to the fact that the internal heat conductivities of the materials were determined under favorable conditions in testing laboratories; that some unforeseen operating condition may occur in the plant; that some extra amount of heat transference should be allowed as a factor of safety in order to be sure that the refrigerating plant will, at all times, maintain the proper temperatures.

Insulating Efficiency.—From the foregoing, it will be observed that the heat that flows through an insulating material depends mostly upon the internal thermal conductivity of the material, and that the resistance to the flow of heat at the surface only slightly reduces the heat transfer. This may be more thoroughly appreciated when the

heat conducted through the wall in Fig. 100 by internal conductivity only, is calculated. This is as follows:

$$K = \frac{1}{\frac{13}{5} + \frac{4}{0.308} + \frac{1}{6.25}}$$
$$= 0.0635 \text{ Btu.}$$

The per cent of increase of heat flow due to neglecting the surface effects is equal to:

$$\frac{0.0635 - 0.0592}{0.0592} = 0.0725 = 7.25\%$$

In general, the magnitude of the value of the internal conductivity of various insulating materials depends upon the structure and density of the material. The conductivity of air is quite low, so that a material containing a large amount of "dead air" will transfer a minimum amount of heat. A material having a large number of cells containing entrapped air would have a high insulating value. From this it will be noted that insulating materials may be judged roughly by noting their densities.

The particular structure of a material also affects its heat transferring qualities. Thus a material made up of many small fibers will conduct more heat along the fibers than across the fibers. This would indicate that fibrous materials would have the lower insulating efficiency.

Choice of Insulation.—The choice of a suitable and efficient insulating material depends upon many considerations. An insulating material has certain definite functions to perform and must be selected with the view of retarding the most heat with the least expenditure of money for a given period of years. Under general conditions, the following are the most important considerations entering into the selection of insulation:

1. Thermal conductivity
2. Durability
3. Sanitation
4. Relative thickness
5. Structural strength
6. Fireproofness
7. Cost

Conductivity.—In order for a material to come into universal application, it must transmit heat at a very slow rate; in short, it should be a good non-conductor of heat. Furthermore, it will be noted that

the loss of heat through a poorly insulated wall is a continual one, while the reduction of the loss to a minimum by the use of a good insulating material of proper thickness is a question of the initial cost of the installation.

Also, when comparing insulating material, it should be noted that the value of the internal thermal conductivity should be used as a criterion of comparison, and the surface effects may be eliminated since all materials have surfaces and hence are on an equal basis in this respect.

Durability.—The durability of insulating materials depends upon the life of the materials used in manufacturing the insulation, the mode of manufacture, and the waterproofness. It is obvious that the materials used, together with the mode of making such materials, should not affect the life of the insulation in respect to making it shorter than the life of the walls.

On the other hand, it is apparent that the materials should be waterproof in order to be durable. The absorption of moisture by capillary attraction and the extraction of moisture from the air by difference of water vapor pressure, would soon fill the insulation with moisture. The presence of water not only increases the conductivity of the material, but also the repeated freezings and thawings of the water result in actual disintegration of the material.

Sanitation.—Since ordinary food products, etc., may be stored in the insulated rooms, it is evident that the insulation should be perfectly sanitary and free from mold, rot, or odor, during the time that it is first installed or any subsequent time. A desirable insulation should be of such a nature as to exclude rats, mice, or other vermin. The interior surfaces should be so constructed that they may be washed with water without affecting the insulation.

Relative Thickness.—The relative thickness or compactness of an insulating material affects the ease of erection and the saving of space which may be used for the storing of commodities. A material that is compact and strong may be shipped, handled and installed with ease. The importance of using a comparatively thin wall of good insulation will be realized when one considers the value of the additional space made available.

As an example in the saving of space, the relative thickness of insulations of the same heat retarding abilities, such as corkboard and wood boards and air spaces, may be considered. Ordinarily, an insulation composed of air spaces and boards will be two to three times thicker than an insulation composed of corkboard. Thus, a wall made up of twelve 1-in. boards and five 1-in. air spaces, having

a thickness of 15.5 in., will have approximately the same resistance to the flow of heat as a wall comprised of 6-in. corkboard having an overall thickness of 7 in.

Structural Strength.—The structural strength of an insulating material determines, to a large extent, its commercial application. The insulation should have sufficient strength to allow it to be installed by ordinary building methods and by ordinary workmen. It should be compact and structurally strong so that it may be incorporated into walls, ceilings and floors of any type of construction. It should have a certain compression strength and should be capable of being nailed, cemented or fastened by any other means into the various locations.

Fireproofness.—A desirable insulator should lend itself to fireproof construction of buildings. It should be slow burning, or fire retarding, and when properly erected it would secure the more favorable insurance rates. An insulation to be slow burning should not support a flame unless it is supplied with heat from an outside source. An insulation is classed as fire retarding when it may be built into solid and compact walls which have no air spaces to act as flues in case of a fire.

Cost.—The total yearly cost of insulation for rooms, pipes, coolers, etc., should be kept down to a minimum. The total cost of insulation depends upon such factors as, initial cost of insulation, fixed charges due to interest, depreciation, repairs, etc., the value of space, the cost of producing refrigeration, etc.

An insulation which possesses the necessary characteristics as previously enumerated and which has a medium total yearly cost, as indicated above, will find universal use.

Types of Insulating Materials.—There are many materials which are used to resist the flow of heat, as may be seen by an inspection of Table 57. The most commonly used materials are corkboard, granulated cork, Fiberglas, redwood bark, mineral wool boards, loose mineral wool, sawdust and shavings, felts and quilts, etc.

Cork.—Cork is the outer layer of the bark of an evergreen species of oak tree. After being removed from the tree and after being dried, it is used to make various articles. The shavings and small particles are collected. This granulated cork is pressed into metal molds and baked at a moderate temperature. The baking process brings out the waterproof gum or rosin in the cork, cementing the whole mass together firmly. After being baked, the boards are trimmed to size.

Corkboards may also be manufactured in another manner. In this process, pure granulated cork is thoroughly coated with an odorless

and waterproof binding material. It is then pressed into a mold and slightly heated.

The boards are 12 in. by 36 in., and are ¼ to 6 in. thick. The commercial boards weigh approximately 7 to 11 lbs. per cu. ft.

The ability of corkboard to retard the flow of heat is easily understood when the structure of the material is inspected. It is simply a homogeneous mass of tiny air cells separated by thin walls of tissue. Each of these cells contains entrapped air and each one is sealed up tightly. Pure cork contains approximately 43 per cent wood fiber and 57 per cent entrapped air by volume. From its structure, one would expect its conductivity to be low. The commercial grade has a conductivity of 0.27 Btu. per hour per deg. per in. thickness per sq. ft.

Cork, being cellular in structure, does not possess capillary attraction, and since each granule is coated with a waterproof gum or waterproof binder, it is water resisting and impervious to the entrance of water. It will not rot, mold, or give off offensive odors. If it is properly erected, it is vermin proof. The surface may be finished with cement plaster which may be washed with water. Because of the low heat conductivity and compactness, a very small amount of space is required for the insulation. It is structurally strong and therefore easily erected. Solid partitions may be constructed. It may be used in floors, under tanks, machinery, etc.

Cork is slow burning, and its construction into building insulation produces a fire retarding condition. Due to improved methods of manufacture, larger factories, and increased production, the initial cost compares favorably with other types of insulating material, when the ultimate cost of the insulation is considered.

Mineral Wool Board.—Waterproof mineral wool boards are made from mineral wool and waterproofing binder. Mineral wool is made from limestone which is melted at about 3000° F. and then is blown into fine fibers by high-pressure steam. This rock wool is mixed with a binder consisting of paraffin wax, bog peat, etc., and then is molded into boards, generally 16 in. x 36 in. and ½-in. to 3-in. thick. It weighs from 17 to 21 lbs. per cu. ft.

The ability of mineral wool board to retard the flow of heat depends upon the structure, which consists of minute cells of air surrounded by fine short fibers of mineral matter and the binder. The air is held in each cell by a thin film of the binder composition which covers the several fibers of mineral matter.

The board generally consists of about 90 per cent rock or mineral wool and about 10 per cent waterproofed peat binder. It contains approximately 83 per cent entrapped air by volume. Its heat conduc-

tivity is 0.346 Btu. per in. thickness per deg. per hour per sq. ft. During the manufacture, the mineral wool is mixed with the waterproofing binder, which serves to make the material fairly waterproof and firm. Since it is made from mineral wool, peat, and parraffin, it is odorless and free from decay and deterioration. It may be plastered with cement mortar which makes a sanitary construction. It is fairly compact and occupies a small amount of space.

The structural strength is somewhat below that of corkboard. The flying particles of mineral wool make it somewhat disagreeable for workmen to erect. It is slow burning and may be erected into a fire retarding construction. In case of fire, however, the binder may be affected, allowing it to disintegrate into a granular state. The initial cost is below that of corkboard.

Fiberglas.—Molten glass is blown into an enclosed chamber under suitable conditions to produce fine fibers of glass. The product is collected and placed in batts and boards of various thicknesses for use in insulating refrigerators and other inclosures. For structures built in place such as cold storage and freezer rooms the Fiberglas is compressed, treated with a binder and enclosed in an asphalt coating. These rigid slabs of insulation are very much like other board forms in size and shape and are erected with the hot dip asphalt method or with other usual methods.

The insulation is also furnished in pre-formed sections for use on brine pipes and other cold lines. It will not rot or decay and it offers excellent resistence to rodents. The density of the material ranges from 2½ to 9 lbs. per cu. ft. with a thermal conductivity of 0.24 Btu. per sq. ft., per in. thickness, per hour, per degree temperature difference in the lower density range.

Insulite.—Insulite is made entirely from selected wood fibers. In the process of manufacture these fibers are chemically preserved and sterilized then fabricated into large rigid sheets and finally kiln dried. The stock sizes of insulite are ½ in. thick, in 3-ft. and 4-ft. widths by 6-ft., 8-ft., 10-ft. and 12-ft. lengths. It may also be had in any special sizes desired, and any thickness can be furnished by stapling or cementing these sheets together.

It has a thermal conductivity of 0.296 Btu. per square foot, per inch thickness, per hour, per degree difference in temperature, and a tensile strength of 350 pounds per square inch. It is odorless under all conditions and it is claimed will not rot, mold, or support fungus or bacteria growths.

In cold storage work insulite is applied in various ways either in built up block formation or in large ½-in. sheets nailed in position to studding or furring strips with all joints broken. The larger pieces

of insulation are advantageous in reducing labor costs. Any suitable waterproofing material may be used for coating the exterior surfaces. The interior surface may be finished with asphalt or with cement plaster, either of which forms an unusually strong bond.

Practical Considerations.—In the actual application of insulation to rooms, pipes, coolers, tanks, etc., it is apparent that each particular plant must have its own individual consideration and prescription. The proper thickness of insulation to be installed on cold storage walls to help maintain a given temperature in an economical manner depends upon several factors, among which the following are the most important:

1. Character of building
2. Thickness of building walls
3. Temperature to be maintained
4. Locality of plant
5. Nature of goods stored
6. Cost of refrigeration

Generally speaking, the most economic thickness of cork, mineral, and other boards, for average conditions in the United States, will be as follows:

−35° to −50° F	12 in.
−25° to −35° F	10 in.
−20° to −5° F	8 in.
−5° to 5° F	6 in.
5° to 20° F	5 in.
20° to 35° F	4 in.
35° to 45° F	3 in.
45° and up	2 in.

The selection of insulation is a question of economic importance and should be considered from all possible viewpoints. The proper insulation and thickness of same should be selected so as to give the lowest ultimate cost rather than the lowest initial cost.

Heat Transfer in Apparatus.—In the foregoing it will be observed that in order to retard the flow of heat from the region of high temperature into the region of low temperature, a wall composed of a non-conductor of heat was interposed between the two regions. This is the principal function of insulation as used in refrigeration structures.

In refrigeration apparatus, the condition is just reversed; that is, it is desired to have the heat flow as fast as possible from the region of higher temperature to the region of lower temperature. Thus, the apparatus in general will contain two fluids separated by a solid wall.

The principles underlying the heat transfer in apparatus are the same as those for the heat transfer in insulation. In other words, there must be a temperature difference; the direction of heat flow is

always toward the lower temperature; the heat may be transferred by any or all of the three methods of heat transfer; that is, by either radiation, convection or conduction. The fundamental heat transfer law for apparatus is the same as for insulation, and is as follows:

$$H = K \times A \times t_d$$

where H = total heat transfer in Btu. per hour
K = heat transfer coefficient, Btu. per hr. per deg. temp. diff.
A = area of heat transmitting surfaces, in sq. ft.
t_d = mean temp. diff.

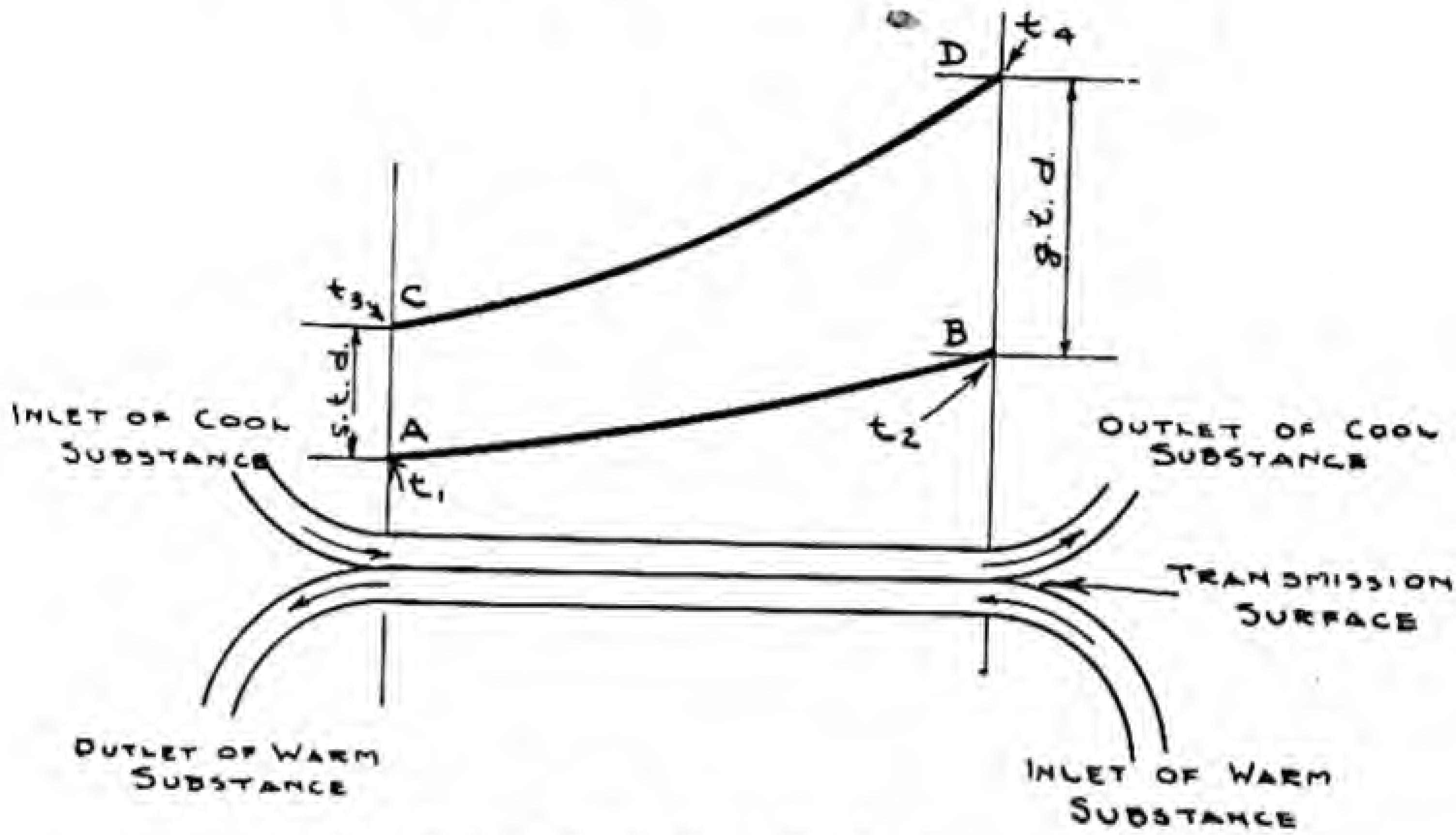


Fig. 103.—Heat Transfer in Apparatus.

In the above, the mean temperature difference, t_d, is the actual mean temperature difference between the separated fluids for the whole period of the heat transfer in degrees Fahrenheit. The heat transfer coefficient, K, is the transfer factor which combines the effects of the three methods of heat transfer into a single constant. Furthermore, it is evident that this heat transfer coefficient takes into consideration other factors such as the properties of the separated fluids, the properties of the separating solid and the effects of the surfaces of the solid.

Mean Temperature Differences.—As previously indicated, in the consideration of insulation, the temperature difference is equal to the arithmetical difference of the temperatures of the separated fluids. It is evident that this is true only when the temperatures of the separated fluids remain constant. If the temperatures of either or both of the separated fluids vary during the heat transferring process, the actual mean temperature difference is a progressive average between the changing temperatures. It is evident that this progressive average cannot be calculated by arithmetic, but must be obtained by means of higher mathematics.

The variation of the temperatures in the apparatus having fluids flowing through it in counter-current flow is shown by Fig. 103. It will be noted that the cool substance is warmed from the temperature at the inlet, t_1, to the temperature at the outlet, t_2, while the warm substance is cooled from the temperature at the inlet, t_4, to the temperature at the outlet, t_3. It is evident that the heat transfer in this case is produced by the average temperature difference between the curves *AB* and *CD*.

This mean temperature difference may be calculated by means of higher mathematics and the result is expressed in the following formula:

$$\text{m.t.d.} = 0.4342 \frac{\text{g.t.d.} - \text{s.t.d.}}{\log. \dfrac{\text{g.t.d.}}{\text{s.t.d.}}}$$

where m.t.d. = mean temp. diff.
g.t.d. = greater temp. diff.
s.t.d. = smaller temp. diff.
0.4342 = a constant
log. = logarithm of

This formula holds true regardless of the direction of the flow of the fluids. The warm and cool fluids may flow in the same direction, which is termed "parallel flow," or they may flow in an opposite direction, which is called "counter-flow."

It will be noted that the above formula for the mean temperature difference involves the use of logarithms. In order to make the labor of solving the formula less, and to make it more easily understood, the above formula has been calculated for a number of conditions. The mean temperature difference in this case may be expressed by the following formula:

$$\text{m.t.d.} = \text{g.t.d.} \times \text{M}$$

where M = coefficient for the mean temp. diff.

The values for the coefficient for the mean temperature difference, M, for the above formula may be taken from Table 59. The coefficient from Table 59 should be taken at a point corresponding to the quotient of the smaller temperature difference divided by the greater temperature difference. As an example, in the above calculations the cooling of water from 80° F. to 40° F. by brine warming from 16° F. to 20° F. will be considered. These temperature variations are shown by Fig. 103. The curve *AB* shows the brine warming from 16° to 20° F., while the curve *CD* shows the water cooling from 80° F. to 40° F. while flowing in opposite direction to the brine.

The greater temperature difference is equal to the temperature at *D*, 80° **F.**, minus the temperatlre at *B*, 20° F., equals 60° F. The smaller temperature difference is equal to 40° F. minus 16° F. equals

24° F. The quotient, $\frac{s.t.d.}{g.t.d.}$ is equal to $\frac{24}{60}$ equals 0.40. From Table 59, the value of the coefficient, M, corresponding to 0.40 is equal to 0.658. The mean temperature difference is found as follows:

$$\text{m.t.d.} = \text{g.t.d.} \times M$$
$$\text{m.t.d.} = 60 \times 0.658$$
$$\text{m.t.d.} = 39.48°$$

It will be noted that the arithmetical temperature difference would be equal to $\frac{60 + 24}{2}$ which equals 42° F. From the above, it will be noted that the value 39.48 is the proper value to be used in calculation and that the false value of 42° F. is 6½ per cent above the actual temperature difference.

TABLE 59.—COEFFICIENTS FOR MEAN TEMPERATURE DIFFERENCES (M).

Smaller Difference / Greater Difference	Coefficient M	Smaller Difference / Greater Difference	Coefficient M
0.0025	0.166	0.2000	0.500
0.0050	0.189	0.2100	0.509
0.0100	0.215	0.2200	0.518
0.0200	0.251	0.2300	0.526
0.0300	0.277	0.2400	0.535
0.0400	0.298	0.2500	0.544
0.0500	0.317	0.3000	0.583
0.0600	0.335	0.3500	0.624
0.0700	0.352	0.4000	0.658
0.0800	0.368	0.4500	0.693
0.0900	0.378	0.5000	0.724
0.1000	0.391	0.5500	0.756
0.1100	0.405	0.6000	0.786
0.1200	0.418	0.6500	0.815
0.1300	0.430	0.7000	0.843
0.1400	0.440	0.7500	0.872
0.1500	0.451	0.8000	0.897
0.1600	0.461	0.8500	0.921
0.1700	0.466	0.9000	0.953
0.1800	0.478	0.9500	0.982
0.1900	0.489	1.0000	1.000

Heat Transfer Coefficients.—The magnitude of the value of the overall heat transfer coefficient depends upon several factors, among which the following are the most important:

1. Velocity of separating fluids
2. Kinds of fluids
3. Density of fluids
4. Temperature of both fluids
5. Kinds of substance in separating wall
6. Thickness of separating wall
7. Character of surface of separating wall
8. Surface effects.

The velocity of either or both of the fluids has a great effect on the overall heat transfer coefficient. Increasing the velocity increases the

coefficient, while decreasing the velocity will decrease the coefficient. Thus, in order for any piece of apparatus to transfer the maximum amount of heat, the velocity should be kept as great as possible.

The exact nature of the separated fluids also affects the heat transfer coefficient considerably. In general, when both fluids are liquids and when the velocity is reasonable the heat transfer is good, but when either or both of the fluids are gases the heat transfer is slow, with the principal resistance to the flow of heat being at the surface in contact with the gas. Increasing the density of the fluids likewise increases the flow of heat. This is particularly true when the fluids are gases.

The rate of heat transfer also seems to depend upon the absolute temperatures of the fluids, together with their relative temperature differences. However, for ordinary refrigeration work the value of the heat transfer coefficient may be assumed to be constant.

The influence of the kind of material used in the separating wall depends upon the conductivity of the same. Thus, a separating wall composed of copper, iron or steel, will conduct the maximum amount of heat. The thickness of the separating wall will have the same effects as before noted in the transfer of heat through insulation.

The character of the surfaces of the separating wall also affects slightly the rate of heat transferred, depending upon whether or not these surfaces are rough, smooth, polished, light or dark. The surface effect is due to the formation of a very thin film of the fluid or foreign matter on the surface of the separating wall. This film may consist of the fluid itself, or some foreign matter as ice, snow, grease, oil, scale, etc.

From a consideration of the above factors, it is quite apparent that it is not possible to determine the magnitude of the heat transfer coefficient by means of calculations. The proper method of determination of these constants is to base the considerations upon experiments and theoretical analysis. In the first place, it is necessary to know in general just what are the factors and how they affect the heat transfer coefficient. After these factors are known, the heat transfer coefficients for all conditions should be determined directly by experiment. Then, as soon as the refrigerating engineer has the coefficient of heat transfer, together with the knowledge of the factors affecting the rate of heat transferred, he is in a position to operate or design intelligently apparatus for the transfer of heat.

The heat transfer coefficients for various types of commercial apparatus operating under various conditions and temperature differences are shown by Table 60. This table shows the rate of heat transfer in Btu. per hr. per sq. ft. per deg. of av. temp. diff.

In reference to the data on the heat transmission of direct expansion coils, as given in Table 60, *old style coils* are those which are made with "hairpin" bends and in which the liquid refrigerant is admitted to the bottom of the coils and the gas is removed from the top of the coil. *Flooded coils* refers to flooded old style coils used in connection with accumulators. *Flooded liquid recirculating coils* refers to those coils which are supplied with an excess of recirculating liquid ammonia, having short travel of gas and liquid.

The heat transfer coefficients shown in this table are average coefficients; that is, they are for medium frosting of the pipes and for medium collection of foreign matters on the heat transmitting surfaces. For a new apparatus, these coefficients would probably be

TABLE 60.—HEAT TRANSFER COEFFICIENT K IN BTU. PER HR. PER SQ. FT. PER DEG. TEMP. DIFF.

Ammonia Condenser									
Submerged type	K = 30 to 40								
Atmospheric, parallel flow type	K = 50 to 60								
Atmospheric, counter flow bleeder type	K = 125 to 200								
Atmospheric, flooded type	K = 125 to 200								
Double-pipe type									
Velocity of water, ft.p.m.	100	150	200	250	300	350	400		
	K =110	150	180	205	225	240	250		
Shell-and-tube type (Vertical single-pass type)									
g.p.m. per sq. ft. of tube surface	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	
	K^1= 70	120	150	170	190	200	210	220	
Shell-and-tube type (Multi-pass type)	K = 150 to 300								
Baudelot Coolers (Direct Expansion)									
Water coolers	K = 60								
Milk coolers	K = 60								
Cream coolers	K = 50								
Oil coolers	K = 10								
Baudelot Coolers (Brine Type)									
Milk coolers	K = 80								
Water coolers	K = 80								
Brine Coolers									
Shell-and-tube type									
Velocity of brine, ft.p.m.	50	100	150	200	250				
	K = 55	80	100	115	125				
Double-pipe type									
Velocity of brine, ft.p.m.	100	150	200	250	300	350	400		
	K^2= 75	85	97.5	110	120	127.5	135		
Brine Coils In Still Air									
Temp. diff. deg. F.	5	10	15	20	25	30			
	K = 1.5	2.3	2.75	3.1	3.3	3.4			
Brine Coils In Moving Air									
Velocity of air, ft.p.m.	200	300	400	500	600	700	800		
	K = 2.9	4.0	5.0	5.9	6.8	7.7	8.6		
Brine Coils in Spray of Liquid	K = 80								
Brine Coils Submerged in Still Liquid	K = 12								
Congealing Tank Coil in Still Brine	K = 8								

TABLE 60.—HEAT TRANSFER COEFFICIENT K IN BTU. PER HR. PER SQ. FT. PER DEG. TEMP. DIFF.—(Continued.)

Direct Expansion Coils (Old Style) in Still Air							
Temp. diff. deg. F.	5	10	15	20	25	30	
K =	1.0	1.75	2.20	2.5	2.7	2.8	
Direct Expansion Coils (Old Style) in Moving Air							
Velocity of air, ft.p.m.	200	300	400	500	600	700	800
K =	2.3	3.3	4.2	5.0	5.7	6.5	7.0
Direct Expansion Coils (Old Style) in Spray of Liquid... K =	60						
Direct Expansion Coil (Old Style) in Still Liquid........ K =	10						
Direct Expansion Coil (Old Style) in Circulated Liquids							
Velocity of liquid, ft.p.m.	20	25	30	35	40		
K[3] =	13.4	15.0	16.4	17.8	19.0		
Direct Expansion Coils (Flooded) in Still Air							
Temp. diff. deg. F.	5	10	15	20	25	30	
K =	1.3	2.3	2.9	3.3	3.6	3.75	
Direct Expansion Coils (Flooded) in Moving Air							
Velocity of air, ft.p.m.	200	300	400	500	600	700	800
K =	3.3	4.5	5.6	6.6	7.7	8.6	9.6
Direct Expansion Coils (Flooded) in Spray of Liquid.... K =	80						
Direct Expansion Coil (Flooded) in Still Liquid........ K =	13						
Direct Expansion Coil (Flooded) in Circulated Liquid							
Velocity of liquid in ft.p.m.	20	25	30	35	40		
K[4] =	17.9	20	21.9	23.7	25.3		
Direct Expansion Coils (Flooded Liquid Recirculating) In Still Air							
Temp. diff. deg. F.	5	10	15	20	25	30	
K =	1.5	2.6	3.3	3.7	3.9	4.0	
Direct Expansion Coils (Flooded Liquid Recirculating) in Moving Air							
Velocity of air in ft.p.m.	200	300	400	500	600	700	800
K =	3.6	5.0	6.3	7.5	8.6	9.6	10.5
Direct Expansion Coils (Flooded Liquid Recirculating) in Spray of Liquid.................................. K =	90						
Direct Expansion Coils (Flooded Liquid Recirculating) in Still Liquid .... K =	15						
Direct Expansion Coils (Flooded Liquid Recirculating) in Circulating Liquid							
Velocity of liquid, ft.p.m.	25	50	75	100	125	150	
K[5] =	55	70	85	95	105	115	
Double-pipe Exchanger .... K =	80						
De-superheater (Shell-and-tube type) .... K =	25						
Fin Evaporators (6-in. x 6-in. copper fins on ½-in. spacing)							
Temp. diff. between air and suction vapor	5	10	15	20	25	30	
K[6] =	1.87	1.48	1.25	1.09	0.98	0.93	
Fin Evaporators (6-in. x 6-in. copper fins on 1-in. spacing)							
Temp. diff. between air and suction vapor	5	10	15	20	25	30	
K[6] =	3.22	2.68	2.39	2.11	1.92	1.73	
Fin Evaporators (6-in. x 6-in. copper fins on 1½-in. spacing)							
Temp. diff. between air and suction vapor	5	10	15	20	25	30	
K[6] =	3.22	2.68	2.39	2.11	1.92	1.73	

TABLE 60.—HEAT TRANSFER COEFFICIENT K IN BTU. PER HR. PER SQ. FT. PER DEG. TEMP. DIFF.—(Concluded.)

Fin Evaporators (13-in. x 13-in.[7] copper fins on 1-in. spacing) Temp. diff. between air and suction vapor	5	10	15	20	25	30	
	$K^6 =$ 2.60	2.18	1.94	1.69	1.55	1.39	
Fin Evaporators (13-in. x 13-in.[7] copper fins on 1½-in. spacing) Temp. diff. between air and suction vapor	5	10	15	20	25	30	
	$K^6 =$ 3.61	2.99	2.66	2.32	2.09	1.92	
Fin Evaporators (Cast Type[8]) in Moving Air Velocity of air, ft.p.m.	200	300	400	500	600	700	800
	K = 3.4	4.3	5.2	6.0	6.8	7.5	8.1
Holdover Brine Tank Temp. diff. between brine and air	5	10	15	20	25	30	
	K = 0.5	0.83	1.0	1.25	1.33	1.41	
Milk Cooler (Shell-and-tube type,) Velocity of milk, ft.p.m.	50	75	100	125	150		
	K = 90	112	125	135	142		

1. Based on data by Horne and Jenks.
2. Based on tests made by York Mfg. Co.
3. Based on formula $K = 3\sqrt{V}$ where V = Velocity of liquid over coils in ft.p.m.
4. Based on formula $K = 4\sqrt{V}$ where V = Velocity of liquid over coil in ft.p.m.
5. Based on formula $K = 15 + 8\sqrt{V}$ where V = Velocity of liquid over coil in ft.p.m.
6. Based on tests made by Geo. B. Bright Co. on McCord evaporators using methyl chloride. Sulphur dioxide showed approximately 15 per cent small values of K. The value of K is based on total area computed from external area of tubing and both sides of fin surface.
7. This evaporator was made up by using four 6-in. x 6 in. fin plates spaced 1 in. between edges.
8. 4-in. x 7-in. cast fins around a central core. Centers of fins about ¼ in.
9. Based on the use of four 1¼-in. milk tubes in 4-in. shell containing the liquid ammonia.

somewhat larger, while for an older apparatus they may be somewhat smaller.

As an example in the use of these coefficients, it may be desired to determine the amount of direct expansion pipe surface to be installed in a room where the temperature is to be maintained at 30° F. with the temperature of the boiling ammonia in the coils at 10° F., to remove heat at the rate of one ton of refrigeration per day. From the fundamental heat transfer law, H = K × A × td:

$$\frac{288{,}000}{24} = 2.5 \times A \times (30^\circ - 10^\circ)$$

$$\text{therefore } A = \frac{12{,}000}{2.5 \times 20}$$

$$A = 240 \text{ sq. ft.}$$

The lineal feet of pipe may be determined when the size of pipe to be used is known. Thus, if 1¼-in. pipe is to be used, the length

of pipe containing one sq. ft. of external surface equals 2.301 ft. The total lineal feet of pipe to transfer this heat under these conditions would be equal to 240 $\times$ 2.301, which equals 552 ft.

QUESTIONS ON CHAPTER VIII.

1. State and explain the fundamental heat transfer law.

2. Explain how heat may be conducted from a region of high temperature through a material to a region of low temperature.

3. In considering insulation, where is the principal resistance to the flow of heat?

4. How is the overall heat transfer coefficient for an insulating material determined?

5. Calculate the heat that would flow per square foot of area per hour through the wall shown in Fig. 102, when the outside temperature is 80° F. and the inside temperature is 30° F.

6. How much heat would be transmitted through the wall shown in Fig. 102 due to the internal conductivity alone?

7. Describe the various functions of an efficient insulating material.

8. Why is corkboard a good insulator?

9. Explain fully how to obtain the true temperature difference between substances having varying temperatures.

10. Submerged direct expansion coils containing ammonia at 10° F. are used for cooling water from 70° to 40° F. How many feet of 1¼-in. direct expansion coils would be required to cool 500 gals. per hr. under this condition?

CHAPTER IX.

METHODS OF DISTRIBUTION OF REFRIGERATION.

Application of Mechanical Refrigeration.—Having studied the fundamental theoretical principles underlying the action of the various mechanical refrigeration apparatus and structures, and having observed the types and constructions of the various apparatus used for producing refrigeration, it is now an opportune time to notice and study the different systems which are used to transmit the mechanical refrigeration from the points of production to the points of usage in actual industrial plants. In general, the particular method of distribution to be used for a given refrigerating plant will depend upon the local conditions, the size and kind of plant, the nature of the goods stored, refrigeration service requirements, etc.

Distribution Systems.—There are several methods of applying or transmitting refrigeration, or arranging for the absorption of the necessary heat, in actual refrigerating plants. Ammonia or any other volatile refrigerant may be evaporated in a pipe which is placed in the room to be cooled; this is called the "direct expansion" or evaporation of the refrigerant. Refrigeration is produced due to the fact that the refrigerant absorbs its latent heat of evaporation from the material to be cooled. This method of application may be used in freezing rooms, freezer storage rooms, small cold storage rooms, rooms of even temperature, and rooms where loss due to leakage of the refrigerant would be low.

In the brine system cold brine is circulated through a series of pipes or other devices in the room. This brine has been previously cooled by direct expansion or evaporation of ammonia, or other suitable refrigerant. The brine rises a few degrees in temperature as it passes through the materials or space to be cooled, thereby producing the required amount of refrigeration. This system may be used in rooms of uneven temperature, rooms where loss due to leakage of the refrigerant would be great, and large cold storage rooms.

A forced air circulation system may be used also. In this system the refrigeration is accomplished by the circulation of air which has been cooled by the refrigerating machine. In a similar manner the cooled air rises a few degrees when it passes through the materials or spaces to be cooled, thereby producing the necessary amount of refrigeration. The air is cooled by means of a series of pipe coils containing either brine, ammonia or other refrigerant, which are placed in a bunker room. The air is circulated by means of a fan and may be distributed to the room by a duct system. This system may be used on large cold storage rooms in which the temperatures are above the freezing point, on rooms in which excessive ventilation is required, etc.

In small refrigerated rooms or spaces, where the refrigerating machine is designed to operate only a part of the day, the coils are supplemented by "holdover" tanks which are placed in the room. Part of the refrigerating coils are placed in the tank, which contains brine and the other part of the coils are exposed directly in the room. Thus, during the operating period the refrigerating machine cools the room and the brine and during the shut-down period it maintains the temperature fairly uniform.

Direct Expansion Systems.—As previously indicated in the foregoing chapters, volatile refrigerants such as ammonia, carbon dioxide, sulphur dioxide, etc., are circulated through pipes, coils, and other heat transmitting apparatus, for the purpose of producing refrigeration by the direct evaporation of the refrigerants. The temperatures of the volatile refrigerants are maintained a few degrees below the surrounding media which are to be cooled. The heat in the media to be cooled then flows by natural tendency into the boiling refrigerants. The resultant vapor is then withdrawn from the evaporating apparatus by the action of the compressor or the absorber, and a proper amount of liquid is admitted to the evaporating apparatus by means of the expansion or regulating valves, so that the cooling effect is continuous. This method is termed the "direct expansion" or evaporation system.

The evaporating apparatus may consist of a series of pipe coils or conduits in air, gases, or vapors to be refrigerated; pipe coils or conduits in liquids to be cooled; double or triple pipe coolers; shell-and-tube, shell-and-coil, or any other convenient form of heat transmitting surface. In all cases, the refrigerant must be confined and evaporated in a suitable metallic container, which interposes a separating wall between the refrigerants and the media to be refrigerated.

Direct Expansion Pipes in Air.—One of the most common applications of the direct expansion system is the cooling of air in cold storage rooms or other refrigerated compartments by means of pipe coils. In this case the air in the compartment or rooms carries the heat from the

materials, lights, motors, the heat transmitted by insulation, etc., to the cold surface of the pipes. Thus, the pipe coils should be arranged so as to promote a rapid circulation of the air in the room, across the coil surface. The primary function of the circulation of the air is the distribution of refrigeration, and a secondary purpose of the circula-

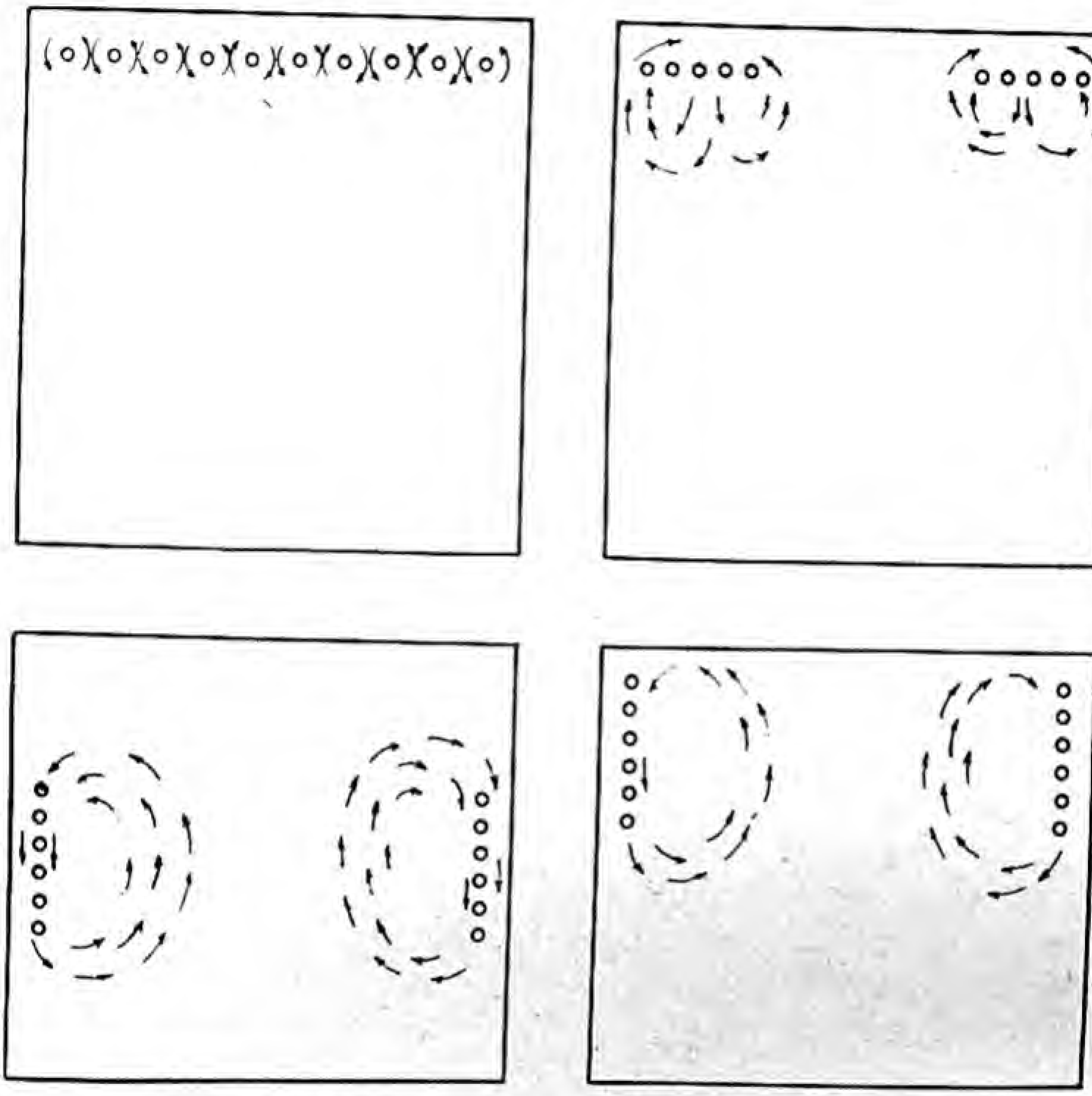


Fig. 104.—Disposition of Pipe Coils.

tion is to equalize the temperature in the room and to purify the air by depositing moisture containing odors, etc., on the cold freezing surface of the coil. The moisture may be deposited on the walls and goods if the refrigeration is produced by the radiation effect rather than the convection effect. The placing of screens or aprons in front of the coil will cut down the radiation effect and also promote circulation. Probably, the best method is to place the coils in a bunker loft which has baffles, and which, together with the room walls, produce a vigorous circulation of the air. This disposition of coils in a bunker gives the best circulation of any gravity air system.

The effect of the disposition of the pipe coils in the room, the loca-

tion of aprons and baffles, etc., may be ascertained by carefully inspecting Figs. 104 and 105. The air is caused to circulate on account of the different densities at the different temperatures. The air, upon passing over the refrigerating pipes, becomes cooled and thereby increases in weight per unit volume. This heavier air tends to sink to-

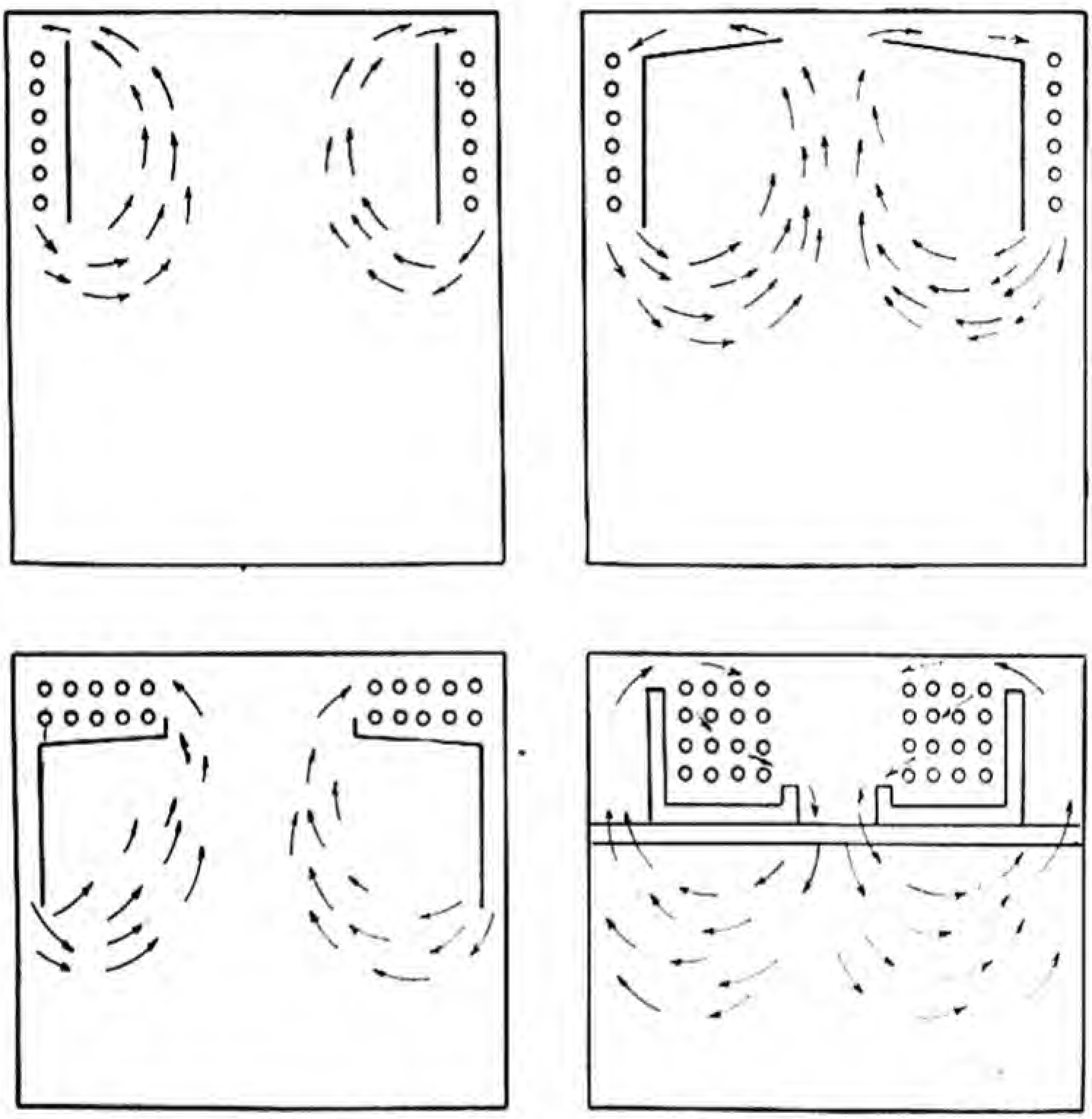


Fig. 105.—Disposition of Pipe Coils, Aprons and Bunkers.

ward the floor, due to the effect of gravity, and, in this manner, displaces the warmer and lighter air, which then flows to the cool surface of the pipe coil. The air, in its circuit about the rooms, rises a few degrees in temperature, probably 2° to 5° F., by absorbing heat from the materials.

The aprons and baffles effect a more even distribution of the air about the room, and they also tend to promote rapid air circulation, due to the chimney effect of the walls of the room and the baffles. Ample area for air passage between the walls and baffles must be allowed, as well as arranging the material to be stored so that the air may circulate freely about the room.

The effect of placing the coils of pipe on the walls and ceiling, or in the bunkers, the effect of the aprons and baffle walls, etc., are shown graphically by Figs. 104 and 105.

The following are some of the major considerations of gravity air circulation in cold storage rooms. This method of distribution of refrigeration tends to purify the air in the room, due to the fact that the moisture which contains objectionable foreign matter is deposited on the cold surfaces of the coils. Generally, some method of removing the ice and frost from the coils, or the water which is condensed by the coils, must be adopted. From the foregoing it will be observed that flat ceiling coils produce very little circulation in the room; that flat coils on the side walls produce fair circulation, and that ceiling coils with baffles produce the best circulation. If baffles or aprons are not provided for the coils, material placed near the coils may be frozen. In general, it will be noted that it is nearly impossible, by the gravity air circulation system, to secure even temperatures in all parts of the room. Furthermore, as previously indicated, the use of this method should be restricted to freezing rooms, freezer storage rooms, small cold storage rooms, etc.

Amount of Direct Expansion Piping Required.—The amount of direct expansion piping to be installed to refrigerate a given room or space depends upon the relative amount of refrigeration that must be produced and the mean temperature difference between the air in the room and the boiling refrigerant. The magnitude of the temperature difference to be used depends upon economic considerations, such as the cost of producing refrigeration, cost of refrigeration equipment, cost of insulation, relative size of plant, etc.

The temperature difference also varies with the relative temperature of the room, being greater on the rooms of higher temperature and lower on rooms of lower temperature. This is due to the fact that, as the temperature of the room, and hence the boiling temperature of the refrigerant, is lowered, the power required for compressing the ammonia increases rapidly. (This condition in part accounts for the use of the absorption machine, or the two-stage compression machine for low temperature refrigeration.)

The larger amount of pipe coil surface will produce the more economical operating conditions, since the suction pressure (and temperature) may be carried at a higher point. Thus, for a given amount of heat transfer, the temperature difference is reduced in proportion to the pipe surface. In general, the suction pressure should be carried as high as possible and still maintain the desired results. An additional advantage of higher suction pressures is that the tonnage capacity of the compressor per cubic foot of displacement increases as the suction pressure is increased.

In general, with ammonia as the refrigerant, the following approximate temperature differences may be used between the room and the boiling ammonia:

Room temperature, deg. F	-10	0	10	20	30	40	50	60
Ammonia temperature, deg. F	-25	-15	-5	3	10	16	22	26
Temperature difference, deg. F	15	15	15	17	20	24	28	34

The amount of piping is also affected by the relative magnitude of the mean temperature difference, since, in the gravity circulation of the air, the heat transfer coefficient decreases as the temperature difference is decreased.

The amounts of surface per ton of refrigeration were calculated by means of the standard heat transfer law and are shown in Table 61. These surfaces may be transferred into equivalent lineal feet of pipe by simply multiplying the amount by the length of the particular size of pipe which has an exterior surface of one square foot.

TABLE 61.—DIRECT EXPANSION COIL SURFACE IN SQUARE FEET PER TON OF REFRIGERATION.

Suction Temperature	-10	0	10	20	30	40	50	60
	Room Temperature							
26						570	200	141
22					1000	334	172	127
16					570	200	142	109
10				800	240	160	120	96
3			1150	353	178	130	112	85
-5			400	193	137	107	88	74
-15		400	193	137	107	88	74	64
-25	400	193	137	107	88	74	64	57

The lengths of the different sizes of pipes which have an exterior surface of one square foot may be found in any table of pipe dimensions.

The particular size of direct expansion pipe to be used depends upon the local conditions in the plant. In general, smaller pipes are used in the smaller rooms, while larger pipe are installed in the larger rooms. The length of the individual coils is determined by the ability of the vapor to flow through the pipes without sustaining too large a pressure drop. The following tabulation indicates the maximum amounts of piping that may be fed by one expansion valve, and in most cases smaller amounts should be used:

Pipe Size	Maximum Length of Direct Expansion Coils
¾ in.	900
1 in.	1,100
1¼ in.	1,300
1½ in.	1,500
2 in.	1,900
2½ in.	2,300

Direct Expansion Coils in Liquids.—Much refrigeration is produced by evaporating the liquid refrigerant in pipe coils which are placed directly into the liquids to be cooled. In this respect mention may be made of such applications as coils for brine "holdover" and "congealing" tank, ice tank, water cooling tanks, brine cooling tanks, etc. The question of the disposition of this type of coils is not so important as that concerning the placing of the direct-expansion coils in the air.

Probably the major considerations are those of keeping the coil surfaces free from foreign matter, and the arranging for a rapid flow of the liquids across the coil surfaces.

Holdover Tanks.—Holdover tanks are used to keep storage rooms cold during the period that the refrigerating plant is shut down, or, in other words, they hold over the low temperature of the room from the time the refrigerating plant is stopped until it is started again. They are used principally on small plants which do not operate continuously during all of the day, automatic refrigerating plants, etc.

Holdover tanks are steel tanks which contain a strong solution of brine, and are placed in storage rooms in order to maintain as low a temperature as possible during the period that the compressor is stopped. Direct-expansion coils are immersed in the brine in the holdover tank and thus, by the evaporation of the liquid refrigerant, the heat absorbed by the brine during the shut-down period and the heat that is absorbed by the tank during the operating period is extracted. The brine in this manner is cooled a few degrees during the operating period of the compressor, and is warmed by the absorption of heat during the shut-down period. The heat that is absorbed is shown by a change of temperature, and is thus the sensible heat of the brine.

The amount of heat that will be absorbed by the brine depends upon the weight of the brine, the specific heat, and the temperature range, and may be expressed in a formula as follows:

$$H = W \times S \times (t_1 - t_2)$$

where H = heat absorbed by brine
W = weight of brine
S = specific heat of brine
t_1 = higher temperature
t_2 = lower temperature

The heat absorbed by one cubic foot of brine having a specific gravity of 1.2, a specific heat of 0.7, and temperature range from 6° to 22° F., may be calculated as follows:

$$H = (62.5 \times 1.2) \times 0.7 \times (22 - 6)$$
$$= 840 \text{ Btu. per cu. ft.}$$
$$\text{or } 840 \div 75 = 11.2 \text{ Btu. per lb.}$$

The desirable disposition of the holdover tanks and coils may be seen by an inspection of Fig. 106.

Part of the direct expansion piping is immersed directly in the brine, while the rest is exposed to the air. The coils and tanks should be located in bunkers, as shown, to facilitate the vigorous circulation of the air. The tank coil and the adjacent room coil should be connected in series, and fed by one expansion valve, feeding through the brine tank coil first. The exact arrangement of tanks and coils will depend upon the plant condition, the size of the room, etc.

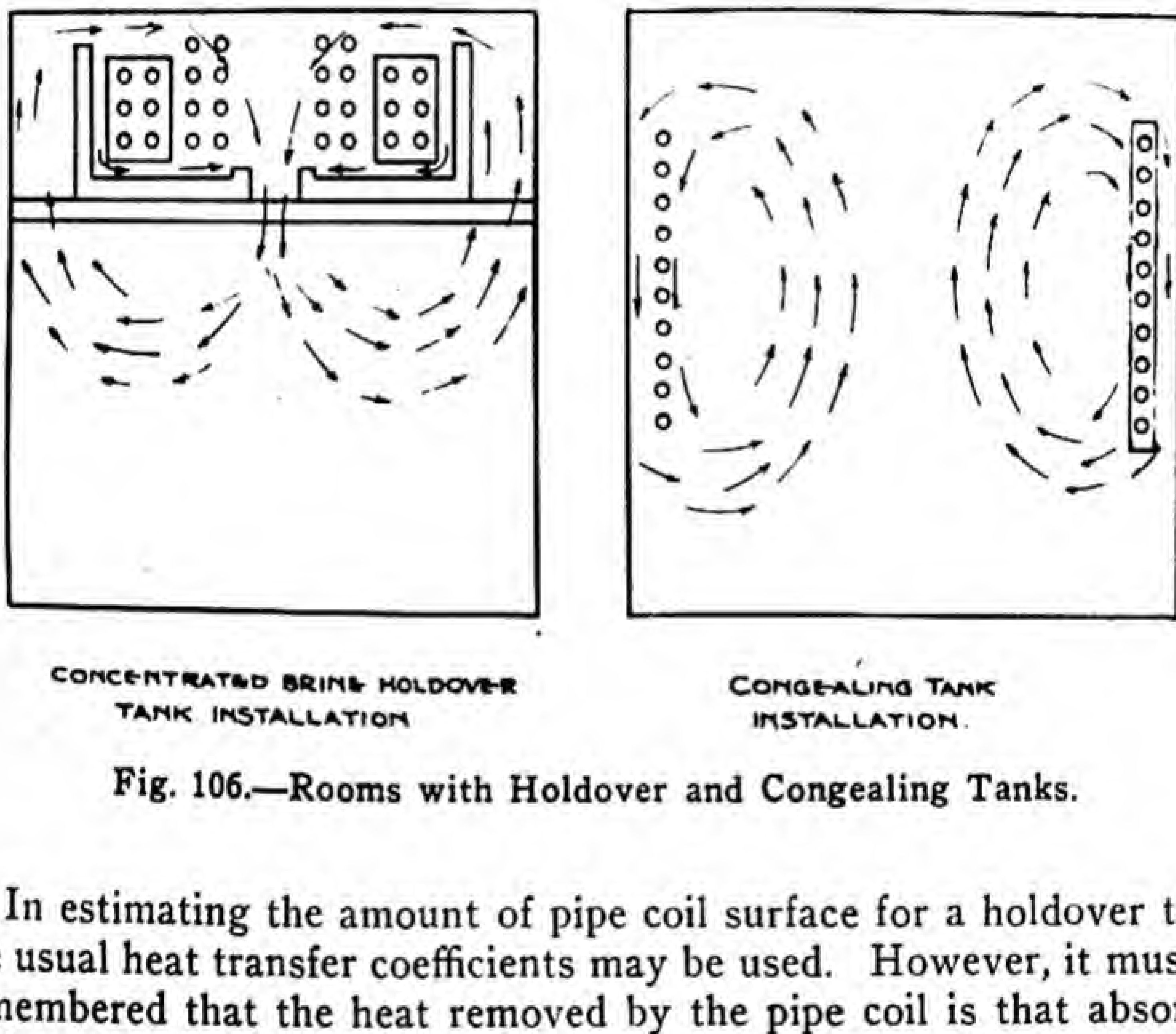


Fig. 106.—Rooms with Holdover and Congealing Tanks.

In estimating the amount of pipe coil surface for a holdover tank, the usual heat transfer coefficients may be used. However, it must be remembered that the heat removed by the pipe coil is that absorbed during the shut-down period, together with that which is absorbed while the refrigerating plant is in operation. Also, it should be remembered that the amount of heat absorbed by the coils exposed to the air is reduced by the amount that is absorbed by the cold holdover tank surfaces.

The size of the holdover tanks depends not only upon the necessary volume of brine that is required, but also upon the amount of heat that must be absorbed from the air by the cold surfaces. In estimating the amount of surface that is required the usual heat transfer coefficients (equivalent to approximately one-half of those of the direct expansion pipes) may be used. The heat transfer coefficient for the holdover tank surface, of course, must be reduced as the temperature difference is reduced.

Congealing Tanks.—Congealing tanks serve the same general purpose as holdover tanks, but they operate on a different principle. They contain a weak solution of brine, which freezes or congeals into a more or less solid mass during the operating period of the refrigerating plant. This type of tank is used on the smaller rooms for the purpose of keeping them cold while the refrigerating plant is not in operation. The tank absorbs the heat that is transmitted by the insulation and any other heat, causing the congealed mass in the tank to melt. The principal part of the refrigeration is produced by the melting of the frozen brine, thus making use of the latent heat of fusion. Advantage may be taken also of the sensible heat of the brine before and after it freezes, due to a temperature change.

Calcium chloride is generally used in congealing tanks, because it is best suited for the purpose. The calcium chloride brine solidifies and then melts in a more suitable manner than common salt or sodium chloride brine. This is due to the fact that calcium chloride does not seem to form such large crystals of the solid substance as sodium chloride, but remains in the form of small crystals which are disposed between the larger crystals of ice. Upon melting, the calcium chloride enters readily into a solution with the water which is formed from the melting ice crystals.

However, common salt or sodium chloride brine may be used instead of calcium chloride, regardless of the fact that it may not give quite as desirable service.

The refrigeration produced by the congealing or frozen brine type of holdover tank may be realized by means of the following calculations. With this type of tank a freezing temperature of 10° to 15° F. may be selected, depending upon the temperature of the room and the temperature of the ammonia in the cooling coil. If a freezing temperature of 10° F. is desired, a solution containing about 16 per cent of dry calcium chloride would be used. As the dry calcium chloride must have water equivalent to one-third of its weight for water of crystallization, the water available for freezing would be as follows:

$$100 - (16 + 0.333 \times 16) = 78.67\%$$

Using a latent heat of fusion of 144 Btu. per pound, the heat that would be available in thawing would be:

$$0.7867 \times 144 = 113.28 \text{ Btu. per lb.}$$

In order to compare the amount of heats that are available for refrigeration in both the concentrated brine holdover tank and the frozen brine type, working between 6° and 22° F., the sensible heat of the brine before and after freezing may be determined as follows, if the

specific heat of the brine is assumed to be 0.75 and that of the frozen brine to be 0.5:

$$(22 - 10) \times 0.75 = 9.0 \text{ Btu.}$$
$$(10 - 6) \times 0.50 = 2.0 \text{ Btu.}$$

Assuming the latent heat of fusion of calcium chloride to be 73 Btu. per pound, the heat required to melt the small amount of calcium chloride would be:

$$0.16 \times 73 = 11.68 \text{ Btu.}$$

The total refrigeration per pound would be:

$$113.28 + 9.0 + 2.0 + 11.68 = 135.96 \text{ Btu.}$$

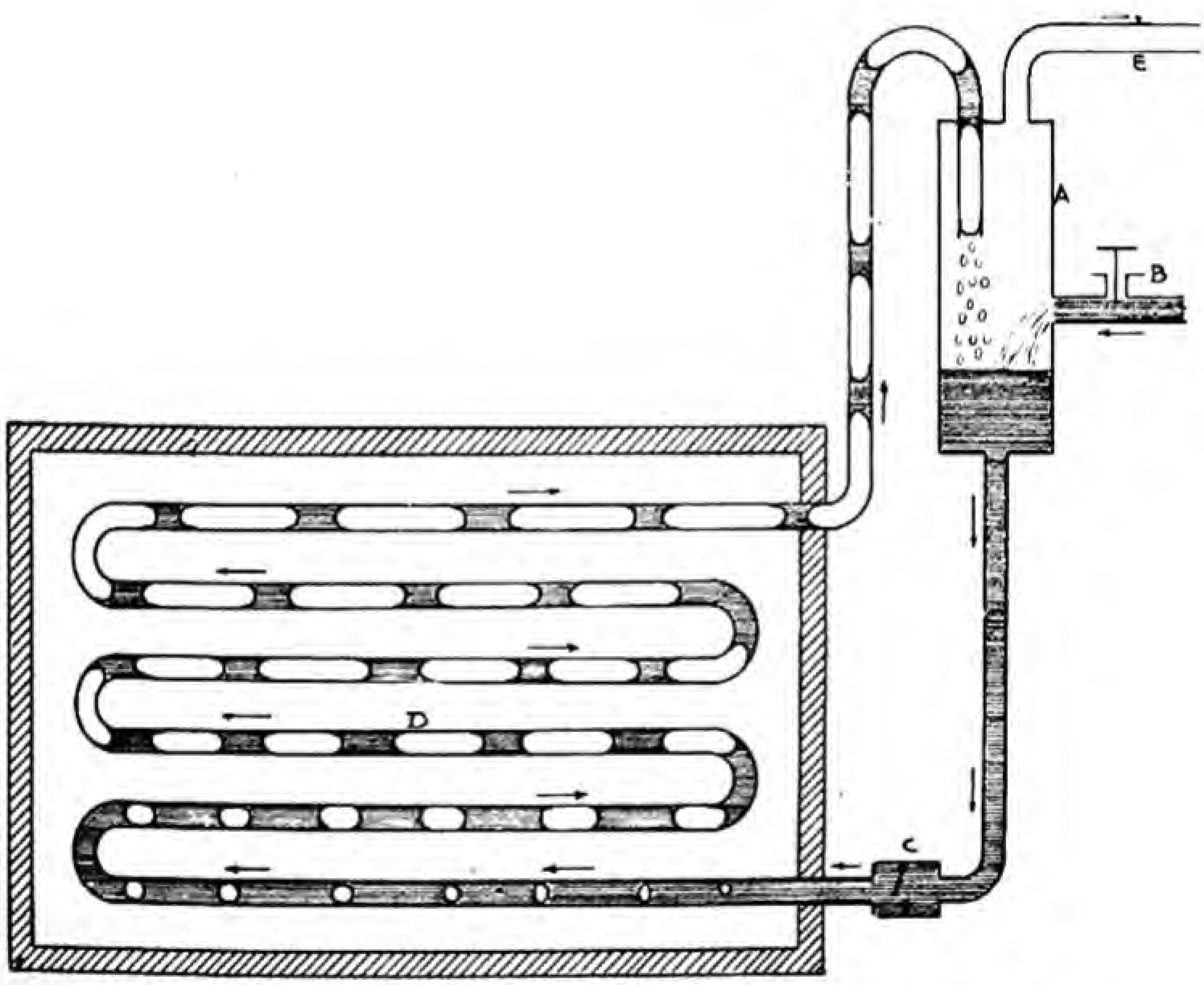


Fig. 107.—Principle of Operation of Flooded Systems.

Comparing this with the amount of refrigeration produced by the concentrated brine type, it will be observed that the frozen brine type has a very much larger cooling capacity for similar conditions. The ratio in this case is $135.96 \div 11.2 = 12.15$. This means that the congealing type tanks are relatively small in volume, as compared with the concentrated brine type. It is evident that the congealing type of tank would require a smaller amount of calcium chloride, and that a lighter tank and room construction could be used.

The arrangement of the direct expansion coils, the tank coil, and the congealing tank in a small cold storage room is shown by Fig. 106.

Other Apparatus for Direct Expansion.—There are many other forms of apparatus which may be used for the direct evaporation of the refrigerant. The forms of apparatus vary with each individual plant and requirement. Such forms as double- and triple-pipe construction, shell-and-tube, shell-and-coil, etc., are the most common. Such apparatus may be used to cool various liquids, vapors, and gases in industrial plants.

Flooded System of Evaporation.—The flooded system is one in which the evaporating surface is partially covered with the liquid refrigerant, as would be implied by the name. Evaporating surfaces such as pipe-coils, double-pipe apparatus, etc., may be operated in this manner. The principle of operation may be ascertained by an inspection of Fig. 107. The apparatus in this figure consists of the following principal parts: an accumulator (A); an expansion valve (B); a check valve (C); an evaporating coil (D); and a suction connection (E), to the refrigerating machine. The warm high pressure liquid refrigerant is admitted to the tank or accumulator (A), through the regulating or expansion valve (B). Since the accumulator (A) is connected to the refrigerating machine through the connection (E), the pressure in the accumulator is very near the suction pressure at the machine. The warm liquid refrigerant admitted by the regulating valve (B) is cooled, therefore, in the accumulator to the saturated temperature corresponding to the suction pressure. The vapor thus formed passes out through the suction connection (E) to the refrigerating machine, while the cool liquid in the bottom of the accumulator is led to the evaporating coil (D) through the liquid line and the check valve (C). The check valve is inserted in the liquid line to insure positive operation of the apparatus, since vapor formed in the lower part of the evaporator may have the tendency to pass back through the liquid connection to the accumulator.

The refrigerant thus enters the evaporating coil as a liquid and contains none of its vapor. As the refrigerant passes through the evaporator it becomes partially evaporized due to the absorption of heat in the refrigerator, and then passes back to the accumulator as a mixture of liquid and vapor. The accumulator is arranged so that it really acts as a separator, allowing the vapor to be taken to the refrigerating machine through the suction connection (E) and permitting the entrained liquid to be deposited in the bottom of the accumulator. The liquid accumulates here, and with that admitted through the regulating valve (B), returns to the evaporator, thus making the action continuous and positive.

It will be noted that the flow of the liquid refrigerant is due to the effect of gravity principally, so that this system of evaporation may be termed the gravity feed system. It will be further noted that the term flooded is used to designate the condition of the interior surfaces of the evaporator, and that strictly speaking the surfaces are not flooded, because the vapor required to produce the refrigerating effect must necessarily be in contact with the surfaces. The action in the flooded evaporator is quite similar to that of a water tube boiler for steam generation. The refrigerant in the evaporator is in the form of a mixture of the liquid and vapor, as indicated by Fig. 107. The expansion valve (B) must be regulated so that the liquid level is maintained at a certain point in the lower part of the accumulator.

It is important to notice the relative advantages and disadvantages of the flooded system, together with the causes for same. The principal advantage is that of simplicity of operation. A multiplicity of pipe-coils and double-pipe coolers may be connected to suitable liquid and suction headers, which are in turn connected to an accumulator, having a single liquid connection. The number of valves to be regulated are thus greatly reduced, making the actual operation of the evaporating system more simple. It regulates automatically the suppply of refrigerant to each section of the evaporator, supplying the refrigerant in proportion to the refrigeration requirement.

In addition, the accumulator eliminates the entrained liquid from the suction vapor, thereby making the plant safe from priming accidents. The vapor returns to the refrigerating machine very nearly saturated and only slightly superheated. In a compression plant, the compressor would then be under the most favorable operating conditions. Also, due to the fact that the interior surfaces of the evaporator are partially covered with liquid, and due to the fact that the velocity of the mixture is quite appreciable, the rate of heat transmission is good. This would mean that a lesser amount of surface may be used, or a higher suction pressure may be employed. The principal disadvantage of this method of evaporation lies in the fact that a large amount of liquid refrigerant must be used to charge the system.

The flooded systems may have different methods of connections as shown by Fig. 108. In the first method, the warm liquid refrigerant is led through a liquid cooling coil, which is placed in the accumulator, and is then admitted to the accumulator through the expansion valve. The vapor formed by the cooling of the liquid goes directly to the refrigerating machine. The precooled liquid then flows by gravity into the evaporator. If the expansion valve is not properly regulated, the vapor leaving the accumulator may become slightly superheated; also, the transfer of heat to the liquid refrigerant in the bottom of the accumulator may cause more or less violent boiling of the refrigerant.

In the second method, the operation is the same as that previously described.

In the third method, the warm liquid refrigerant is led through the liquid precooling coil in the accumulator, after which it is expanded directly into the evaporator. In this system, the liquid is separated from the return vapors and the warm liquid is precooled, but the liquid is not fed by gravity. The warm liquid is cooled by evaporation of the

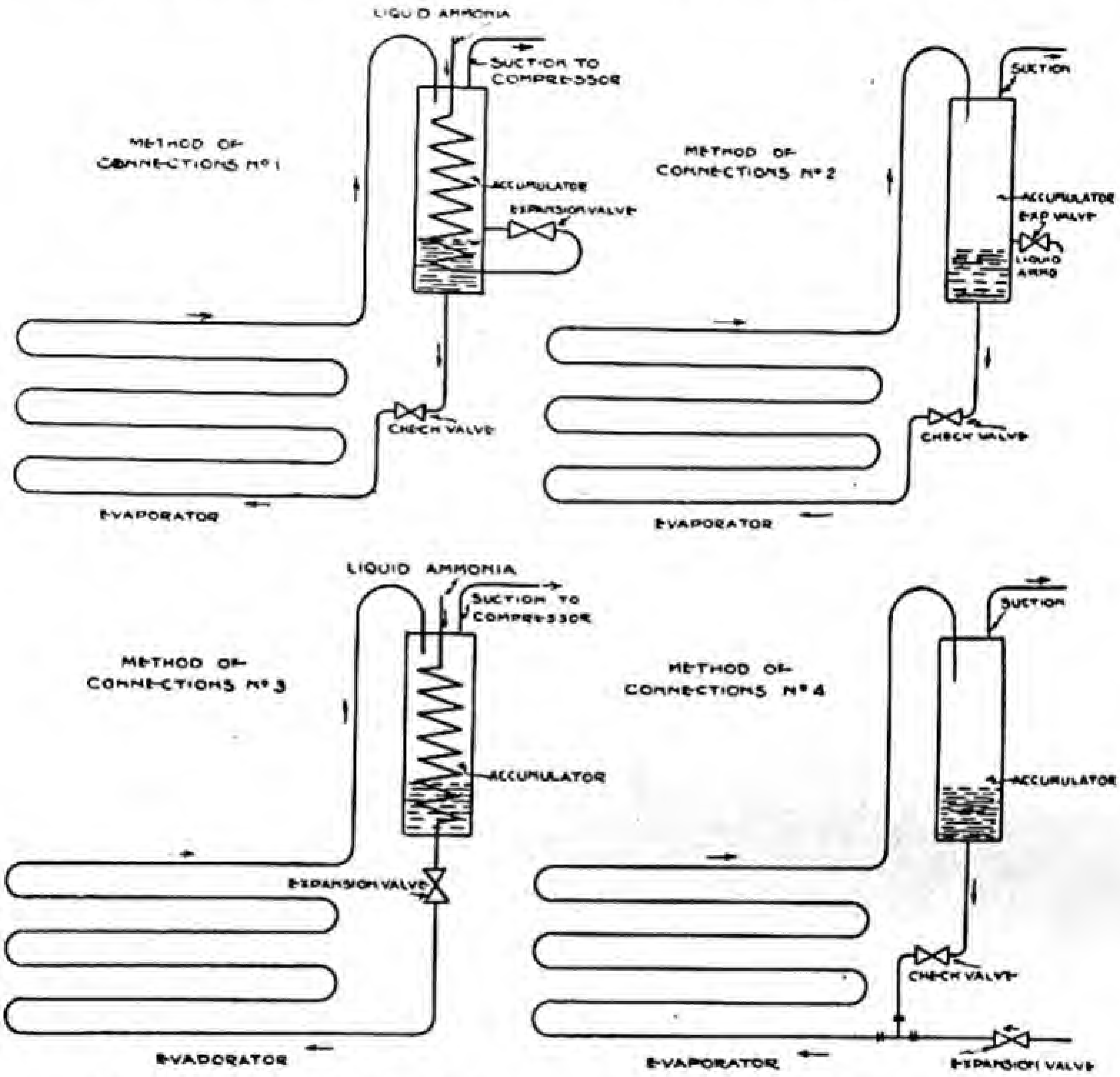


Fig. 108.—Methods of Connections for Flooded Systems.

excess liquid from the evaporator. Unless the expansion valve is properly regulated, superheating of the suction vapor may occur. Also, this method does not utilize the effect of the velocity of the rapid circulation of the refrigerant through the evaporating coil.

In the fourth method, the warm liquid refrigerant is fed directly into the evaporator. The liquid that is separated from the return vapor is fed by gravity back to the evaporator. The vapor formed at the expansion valve due to cooling the liquid to the lower temperature, of course, must pass through the evaporator.

The flooded system may be used to an advantage on low temperature refrigeration work. It is used quite extensively on the direct expansion sharp freezers, ice cream hardening rooms, and other low temperature applications. It is also used on such refrigeration work as ice making, etc.

Mr. Thomas Shipley read a paper entitled "Recent Developments in Evaporating Systems" on April 19, 1928, before the Philadelphia Chapter of the National Association of Practical Refrigerating Engineers, in which he described certain research and test work on a new type of evaporating coil called a "Herringbone Coil."

Fig. 109 shows the design of this coil in which the pipe is cut into short lengths and welded into the headers as shown.

The extensions of the pipe into each of the headers assist in giving freedom of access to and egress from the evaporating pipes.

The cross-section of the herringbone coil in three sections, single, double, and triple pipe-coils, is shown in Fig. 110.

The rate of heat transfer adopted for this design of coil is 100 Btu. per sq. ft. per hour. The records obtained from 36 installations of this type of vertical trunk evaporators show that the average transfer of heat obtained was 127.3 Btu.

The tabulation covering safe commercial requirements of freezing systems embodying this type of coil through the ranges usually called for in ice making plants has been compiled for a heat transfer rate of 100 Btu. per hr. per sq. ft., and the figures shown in Table 62 will be found very useful for the purpose of determining the possibilities of the vertical trunk freezing system of the herringbone design under different brine temperatures and evaporating pressures. The figures enclosed in circles in the tables are conditions which are considered good commercial practice under the A. S. R. E. Standard—that is, at 155 lbs. condensing pressure and 20 lbs. evaporating pressure. The other figures show the possibilities of this freezing system under a sufficient range of conditions to meet any duty ordinarily required in freezing ice.

For the purpose of comparing the efficiency of the herringbone design of freezing system with the design of freezing systems used in a great number of ice plants in the United States the computations shown in Table 63 have been prepared. This table has the same characteristics as Table 62, except it is compiled for freezing systems made up of vertical continuous coils, using a heat transfer of 15 Btu.

An examination of Tables 62 and 63 will disclose the fact that they are exactly alike except the pipe requirements per ton and the brine velocity over the evaporating surface.

The difference of pipe requirements is directly attributable to three conditions: The length of the pipe runs: the method of feeding the

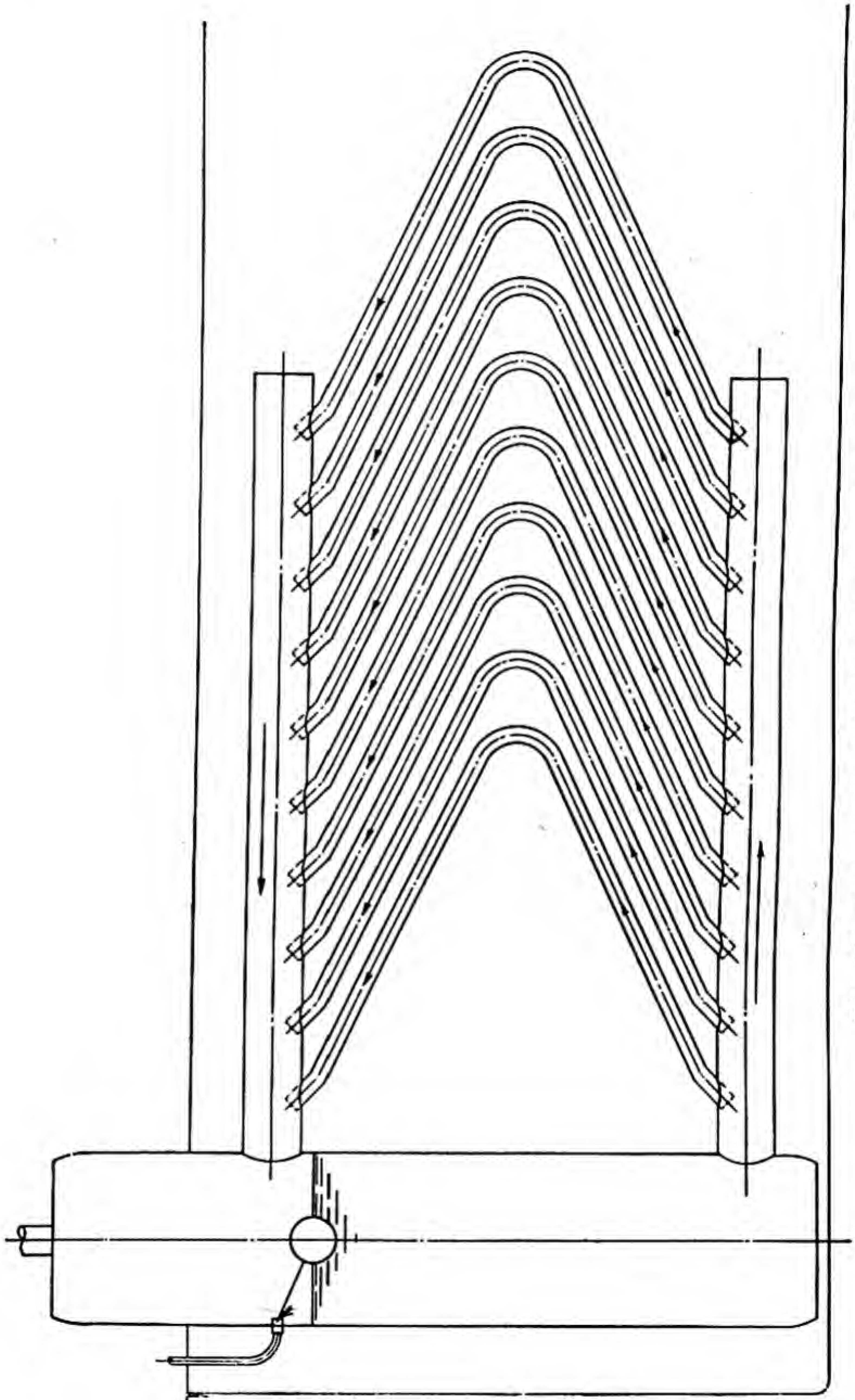


Fig. 109.—Longitudinal Section Herringbone Coil.

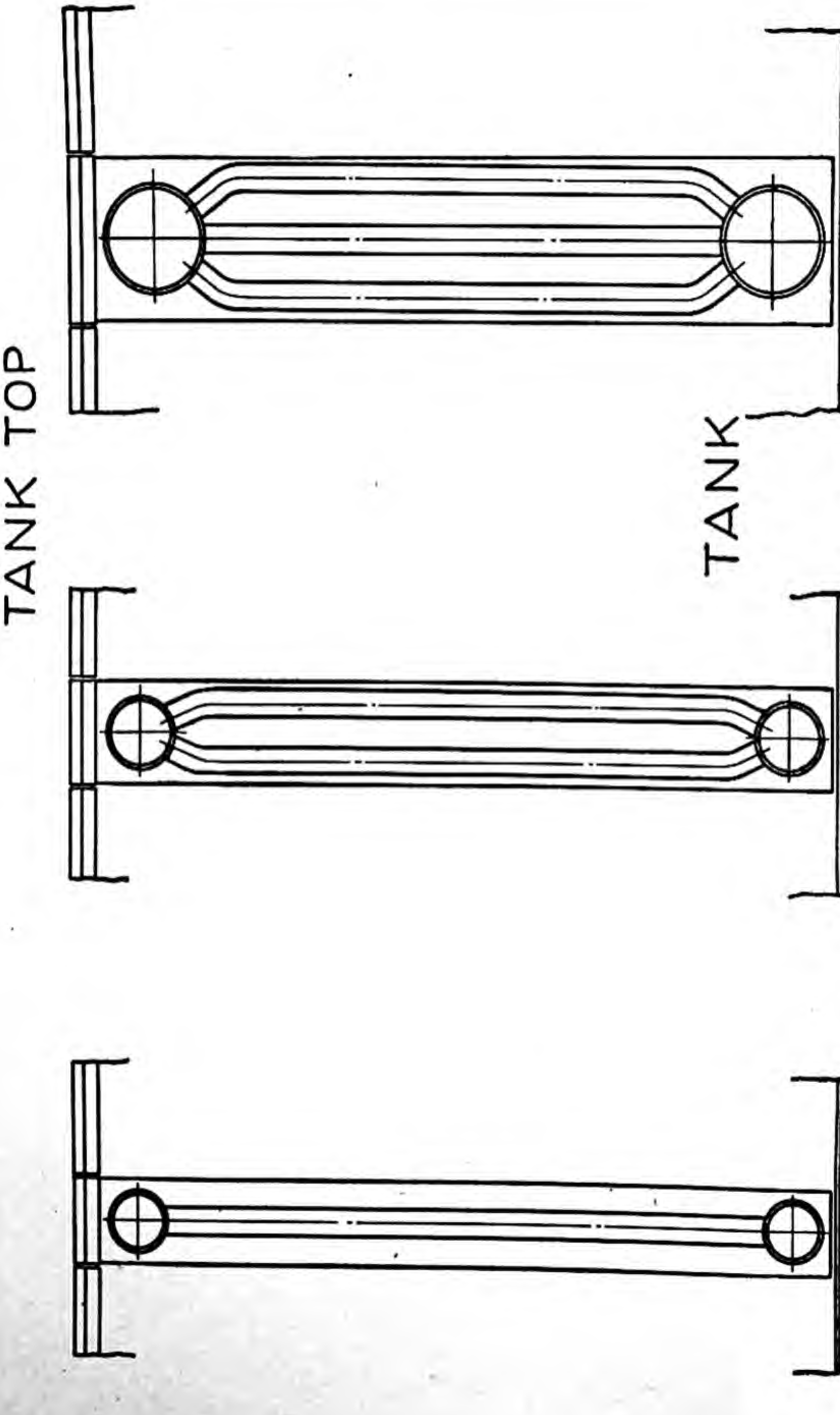


Fig. 110.—Cross-Section Herringbone Coil.

TABLE 62.—VERTICAL TRUNK FREEZING SYSTEMS TABULATION.

YORK ICE MACHINERY CORPORATION, YORK, PA.

VERTICAL TRUNK FREEZING SYSTEMS

Number of 300# Freezing Cans, Suction Pressure and Amount of Evaporating Surface Required in Freezing Tanks With 80° F. Water to Cans

155# CONDENSING PRESSURE.

Brine Velocities { **Through Trunk 150 ft./m.** / **Through Tank 25 ft./m.** }

Approx. Brine Temp. °F.	11" x 22" Cans		11½" x 22½" Cans		Corresponding Temp. °F. Evaporating Pressure = Lbs. G. Lineal Feet of 1¼" Pipe Per Ton of Ice										
	No. Cans per Ton	Freez. Time Hours	No. Cans per Ton	Freez. Time Hours	−1.0° 15#	+0.4° 16#	1.7° 17#	3.0° 18#	4.3° 19#	5.5° 20#	6.7° 21#	7.9° 22#	9.1° 23#	10.2° 24#	11.3° 25#
3.6	7	25.2	7.68	27.6	100.3	144.1	242.8	768.5							
7.1	8	28.8	8.78	31.6	56.9	68.8	85.4	112.5	164.7	288.2	1153.				
9.9	9	32.4	9.87	35.6	42.3	48.5	56.2	66.8	82.4	104.8	144.1	230.6	576.5		
12.1	10	36.0	10.97	39.5	35.2	39.4	44.3	50.7	59.1	69.9	85.4	109.8	153.7	242.8	576.5
13.9	11	39.6	12.07	43.5	30.9	34.2	37.8	42.3	48.	54.9	64.	76.8	96.1	124.6	177.4
15.4	12	43.2	13.16	47.4	28.1	30.7	33.7	37.2	41.5	46.6	53.	61.5	73.2	88.7	112.5
16.7	13	46.8	14.26	51.4	26.0	28.3	30.7	33.7	37.2	41.2	46.1	52.4	60.7	71.	85.4
17.8	14	50.4	15.36	55.3	24.5	26.5	28.6	31.2	34.2	37.5	41.5	46.6	53.	60.6	71.
18.7	15	54.0	16.46	59.2	23.4	25.2	27.1	29.4	32.	34.9	38.4	42.7	48.	54.2	62.3
19.6	16	57.6	17.55	63.2	22.4	24.	25.8	27.8	30.1	32.7	35.8	39.4	43.9	49.1	55.6
20.3	17	61.2	18.65	67.2	21.7	23.2	24.8	26.7	28.8	31.2	33.9	37.2	41.2	45.7	51.2
20.9	18	64.8	19.75	71.1	21.1	22.5	24.	25.8	27.8	29.9	32.5	35.5	39.1	43.1	48.
21.5	19	68.4	20.85	75.1	20.5	21.9	23.3	24.9	26.8	28.8	31.2	33.9	37.2	40.8	45.2
22.04	20	72.0	21.94	79.0	20.	21.3	22.7	24.2	26.	27.9	30.1	32.6	35.6	38.9	42.9
Y. S. A. Ammo. Compr. I. H. P. Per Ton Ice					2.15	2.10	2.04	1.99	1.94	1.89	1.85	1.80	1.76	1.72	1.68
K. W. Hours Per Ton Ice (for 10% Frict. H. P., 90% Motor Eff., 3% Line Loss)					48.6	47.3	46.	44.8	43.7	42.6	41.6	40.6	39.7	38.7	37.7

The figures in circles are our recommendation for A. S. R. E. standard conditions.
Tons Refrigeration per ton Ice 1.67 Heat Transfer 100 B.t.u.

TABLE 63.—DIRECT EXPANSION COIL FREEZING SYSTEMS.

YORK ICE MACHINERY CORPORATION, YORK, PA.

"COIL BETWEEN CANS" FREEZING SYSTEMS

Number of 300# Freezing Cans, Suction Pressure and Amount of Evaporating Surface Required in Freezing Tanks With 80° F. Water to Cans

155# CONDENSING PRESSURE — **Brine Vel. = 25 Ft./Min.**

Approx. Brine Temp °F	11" x 22" Cans		11½" x 22½" Cans		Corresponding Temp. °F. Evaporating Pressure = Lbs. G Lineal Feet of 1¼" Pipe Per Ton of Ice										
	No. Cans per Ton	Freez. Time Hours	No. Cans per Ton	Freez. Time Hours	—1.0° 15#	0.4° 16#	1.7° 17#	3.0° 18#	4.3° 19#	5.5° 20#	6.7° 21#	7.9° 22#	9.1° 23#	10.2° 24#	11.3° 25#
3.6	7	25.2	7.68	27.6	668	960	1617	5120							
7.1	8	28.8	8.78	31.6	379.3	458.7	569.	749.5	1098.	1920.	7685.				
9.9	9	32.4	9.87	35.6	281.9	323.5	374.8	445.3	549.	698.5	960.	1537.	3842.		
12.1	10	36.0	10.97	39.5	234.7	262.7	295.3	337.7	393.8	465.7	569.	731.5	1024.	1617.	3842.
13.9	11	39.6	12.07	43.5	206.2	227.6	252.	281.8	320.2	365.7	426.7	512.5	640.5	831.	1182
15.4	12	43.2	13.16	47.4	187.5	204.8	224.3	247.7	276.8	310.4	353.	409.8	487.6	591.	749.5
16.7	13	46.8	14.26	51.4	173.6	188.5	204.8	224.3	247.7	274.3	307.2	349.2	404.2	472.5	569.
17.8	14	50.4	15.36	55.3	163.3	176.7	190.8	207.8	227.6	249.7	276.8	310.4	353.2	404.2	472.8
18.7	15	54.0	16.46	59.2	156.1	167.8	180.8	195.7	213.3	232.8	256.	284.4	320.	361.4	415.3
19.6	16	57.6	17.55	63.2	149.3	160.	171.6	185.2	200.8	218.	238.2	262.7	292.6	326.8	370.2
20.3	17	61.2	18.65	67.2	144.3	154.5	165.2	177.7	192.	207.8	226.	247.8	274.4	304.3	341.3
20.9	18	64.8	19.75	71.1	140.3	149.8	160.1	171.7	185.1	199.5	216.4	236.3	260.3	287.2	320.1
21.5	19	68.4	20.85	75.1	136.6	145.7	155.2	166.	178.6	192.1	207.7	226.	247.8	272.	301.5
22.04	20	72.0	21.94	79.0	133.3	142.	151.1	161.3	173.2	185.8	200.3	217.3	237.4	259.3	286.
Y. S. A. Ammo. Compr. I. H. P. Per Ton Ice.					2.15	2.10	2.04	1.99	1.94	1.89	1.85	1.80	1.76	1.72	1.68
K. W. Hours Per Ton Ice (for 10% Frict. H. P., 90% Motor Eff., 3% Line Loss)					48.6	47.3	46.	44.8	43.7	42.6	41.6	40.6	39.7	38.7	37.7

The figures in circles are our recommendation for A. S. R. E. standard conditions.
Tons Refrigeration per ton Ice 1.67 Heat Transfer 15 B.t.u.

refrigerant into the coils; the velocity of the brine passing over the surface of the pipe in which the refrigerant is evaporated.

In the herringbone freezing system the length of the pipe runs are from 7½ to 15 ft. for 1¼-in. pipe. The coils are kept as full of the refrigerant as it is possible to do so under the working conditions by float control. The velocity of the brine passing over the evaporating surface is kept as near as possible to 150 ft. per min.

Fig. 111 illustrates another type of flooded coil system. It is called a "paraflow" coil. An accumulator, located at one end of this coil, contains a float control valve. The accumulator is connected to suction and liquid headers as shown. These headers in turn are connected to vertical rows of pipe coils. The liquid ammonia is admitted to the accumulator by the float control valve. From the accumulator the liquid ammonia flows to the liquid header and thence into the vertical coils, by way of a header located at the opposite end of the coil. The ammonia passes through the various pipes of the coil and then into the suction header. From here the gas goes onward to the compressor and excess liquid is returned to the accumulator, where it can again be re-admitted to the liquid supply header.

Brine Distribution Systems.—Refrigeration may be produced in rooms or other compartments by the cooling action of cold brine. The evaporator for the refrigerant cools the brine which is then circulated through the rooms to be cooled. The brine rises a few degrees in temperature, thereby absorbing the necessary amount of heat. The brine may be circulated in the rooms to be cooled by various means.

One of the principal advantages of the brine system is that the compressor may be stopped for a short time, since the brine will maintain a satisfactory temperature for a short period generally. The refrigerant may be isolated in the machine room. However, the brine system is somewhat more expensive to operate, since a lower suction pressure must be used. The brine system also maintains more uniform room temperatures than the direct-expansion system.

Various means have been adopted for distributing the brine in the rooms to be refrigerated. In all of the various methods, the same general considerations relative to air circulation apply to brine piping, as well as to direct-expansion piping. Closed pipe coils are generally used. These are arranged in bunker lofts, or on the ceiling of the rooms, according to the requirements. Sometimes, in packing houses, spray nozzles are used to spray the brine directly into the air to be cooled. These nozzles are arranged in suitable bunker lofts. Other systems, such as the sheet system, the open pan system, the cast-iron section system, etc., may be used.

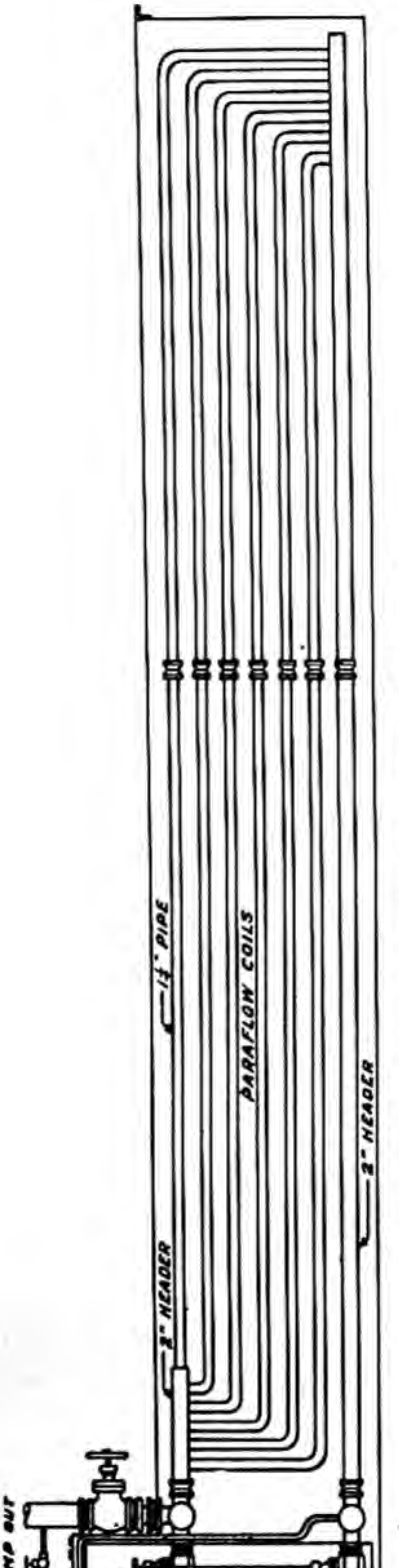


Fig. 111.—Frick Paraflow Coil.

Brine Coils.—Brine pipe coils are arranged in the same general way as direct-expansion coils. Provision should be made at all times for the vigorous circulation of the air. The length of the individual pipe coils depends upon the permissible velocity of the brine and the amount to be circulated. The length, also, would depend upon the local conditions in the plant. Since more brine per ton of refrigeration is circulated generally at low temperatures than at high temperatures, the individual coils are smaller for low temperature work. A coil containing from 100 to 120 ft. of pipe is fed by one feed valve on low temperature work, while a coil containing 400 to 440 ft. may be used per feed valve on high temperature work. The pipe sizes vary from 1¼-in. to 2½-in., the smaller sizes being used on the smaller rooms.

In laying out brine coils and lines, especial attention should be given to the elimination of all unnecessary friction in such coils and lines. The loss due to friction is a continual one, while the installation of sufficiently large pipe is a question of initial cost.

Amount of Brine Piping.—The relative amount of brine piping to be installed to produce a certain amount of refrigeration depends upon the temperature to be maintained and the average temperature of the brine. The magnitude of the temperature difference between the room and the brine and between the brine and the boiling refrigerant depends upon the economic consideration, such as the cost of producing refrigeration, cost of equipment and insulation, relative size of plant, etc. As in the use of direct-expansion pipe, the temperature differences vary with the relative temperature of the room, being smallest at the lower temperatures. The larger amount of brine pipe surface will produce the more economical operating conditions, due to the fact that the suction pressure on the refrigerating machine may be carried at a higher point.

In the event that brine is used for the production of refrigeration, the total temperature difference between the room and the ammonia is somewhat greater than when direct-expansion of the ammonia is used, due to the fact that the brine must be a few degrees below the room temperature, and that in turn, the ammonia temperature must be a few degrees below the brine in order to remove the heat.

The following tabulation shows the variation of the temperature differences with the room temperature:

Room temperature, deg. F	-10	0	10	20	30	40	50
Brine temperature, deg. F	-20	-12	-4	4	12	20	28
Temperature difference, deg. F	10	12	14	16	18	20	22
Room temperature, deg. F	-10	0	10	20	30	40	50
Ammonia temperature, deg. F	-28	-20	-13	-6	1	8	13
Temperature difference, deg. F	18	20	23	26	29	32	37
Brine temperature, deg. F	-20	-12	-4	4	12	20	28
Ammonia temperature, deg. F	-28	-20	-13	-6	1	8	13
Temperature difference, deg. F	8	8	9	10	11	12	15

The following tabulation shows the relative amounts of pipe coil surface that may be used for different room temperatures:

TABLE 64.—BRINE PIPE COIL SURFACE IN SQ. FT. PER TON OF REFRIGERAT ON.

Brine Temperature	Room Temperature						
	-10	0	10	20	30	40	50
28						400	156
20					480	172	115
12				600	212	123	91
4			800	250	132	96	75
-4			344	143	101	78	64
-12		400	156	107	82	66	56
-20	480	172	115	86	69	58	49

The values in the foregoing tabulation are surfaces per ton of refrigeration per day. This surface may be transferred into equivalent lineal feet of any suitable size pipe by multiplying the above amounts by the length of the particular size of pipe which has an exterior surface of one square foot.

Brine coils enclosing the brine circulation are used on coolers, cold storage rooms, freezers, freezer storages, etc. It does not promote an excessive drying action which would affect the amount of moisture in the materials that are stored. Frost and ice tend to accumulate on the cold surfaces, and suitable methods of removing same must be pursued.

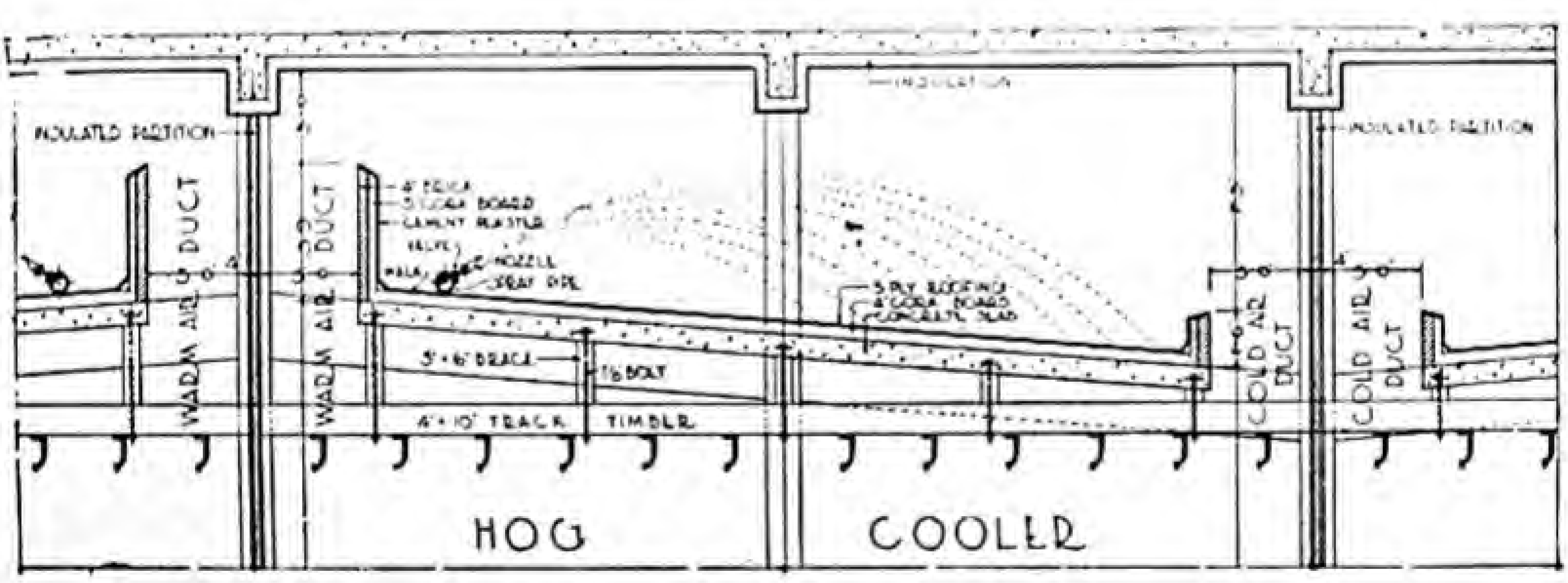


Fig. 112.—Brine Spray Loft.

Brine Spray Nozzles.—Brine may be brought in direct contact with the air to be cooled by spraying it through suitable nozzles, which breaks the brine into small drops. This is termed the open type of brine system, and is used principally in meat chilling rooms in packing houses. The rapid circulation of the air, due to the action of the spray nozzles and the difference of gravities of the cold and warm air, cools the meat in a short time. Moisture from the meat (pork and beef) and

from the air is, of course, absorbed by the brine, which moisture will weaken the brine unless suitable means of concentration are adopted to keep the brine at the proper density.

A typical application of brine spray nozzles to a hog cooler in a packing house is indicated by Fig. 112. The nozzles and the brine supply pipes are located in a bunker loft. The bunkers, with their warm and cold air ducts, are usually made two sections or bays wide, and are insulated with corkboard as shown. The floor of the bunker is made brine-proof by the application of roofing and pitch. The partitions for forming the warm and cold air ducts are insulated to promote air circulation.

The spray nozzles are placed 2 to 5 ft. apart, depending upon the relative size of the nozzles and the amount of brine to be sprayed. The nozzles will require generally an operating pressure of 8 to 15 lbs. per sq. in., being governed principally by the width of the bunker loft. The nozzles have openings varying from ⅛-in. to ¼-in.

The following tabulation gives the capacities of the "Spra-rite" spray nozzles which may be used for spraying brine:

Capacity "Spra-rite" Brine Nozzles

Pipe Conn.	Orifice	Capacity Gallons per Minute—Pounds Pressure			
		8 Lbs.	10 Lbs.	15 Lbs.	20 Lbs.
¼ in.	1/16 in.	....	.50	.60	.70
⅜ in.	⅛ in.	.90	1.0	1.2	1.4
½ in.	3/16 in.	2.0	2.1	2.5	2.9
¾ in.	¼ in.	3.8	4.0	4.7	5.6

The efficiency of the brine spray system depends quite a bit upon the fineness of the brine spray. The nozzles should discharge a uniform, finely divided spray. The system has a very low first cost, and since the pressure is quite low, the cost of operation is minimal. The nozzles, of course, require a certain amount of care and regulation. They must be kept from becoming clogged with foreign matter.

Brine Coolers.—Three principal forms of apparatus are used for cooling the brine in refrigerating plants. These are the coil-and-tank type, the double- and triple-pipe brine coolers, and the shell-and-tube multipass brine coolers.

In the coil-and-tank type an open tank contains submerged pipe coils for the direct expansion of the refrigerant, which is most generally ammonia. The warmer brine is led over these coils and thereby cooled a few degrees before it leaves the tank. The initial cost is greater than that of the other types of coolers, but certain operating risks are reduced. The brine may become weakened, due to the ab-

sorption of moisture, and the result would be that it would freeze at a higher temperature. In the case of an open tank, the only resultant difficulty would be the formation of ice on the evaporator coils, which would reduce their capacity.

In smaller coil and tank brine coolers the only circulation of brine is that due to the main brine pump. This results in a slow movement of the brine over the coils, with the resultant lower heat transfer rate.

In larger coil and tank coolers, the brine is circulated over the coils rapidly by means of motor-driven brine circulators, usually of the propeller type. Such brine tanks are commonly equipped with suitable bulkheads and partitions for directing the brine flow over the coils. This increased brine circulation causes a material increase in heat transfer rate of the coils.

The direct-expansion coils are arranged usually in the form of sections suitable for submerging directly in the brine in the tank. The sections of the coils may be connected to common liquid and suction headers. The coils are usually made of 1-in., 1¼-in. or 2-in. pipe.

In the double- and triple-pipe brine coolers, the refrigerant is evaporated in the annular spaces between the pipe, while the brine flows through the internal pipes. This type of cooler may be used readily on the closed system of circulation, in which no moisture is absorbed by the brine. It is also applicable to the balanced system of circulation. It has the disadvantage of having a multiplicity of joints, and it is open to the danger of the brine becoming frozen in the tubes, if the brine is allowed to become weakened due to the absorption of moisture.

In some cases it is advisable to use an evaporating pressure control device, for the purpose of limiting the pressure in the cooler and thereby protect the cooler from the danger due to freezing of brine.

Double-pipe and triple-pipe brine coolers are usually made in sections which are 12- to 16-pipes high, these sections being connected to common liquid and suction headers. The brine connections are likewise made to common brine return and supply headers.

The double-pipe brine coolers are usually made of 1¼-in. and 2-in. pipe, or 2-in. and 3-in. pipe.

The shell-and-tube brine cooler as illustrated by Fig. 113 consists of a shell which has suitable heads for retaining the tubes. Heads enclose the brine circulation, causing the brine to pass the entire length of the cooler several times. The brine flows through the tubes while the refrigerant is evaporated in the shell. They are open to the danger of the brine becoming frozen in the tubes, but, on the other hand, they have a very small number of joints. They are readily applicable to the closed system of refrigeration.

Shell-and-tube brine coolers of the single-pass type are used in many open systems such as ice tanks, brine tanks, and water cooling

tanks. Usually the brine is forced through the coolers by means of a motor-driven propeller located at one end of the cooler. A typical arrangement of an ice tank using a shell-and-tube cooler is shown in Fig. 114.

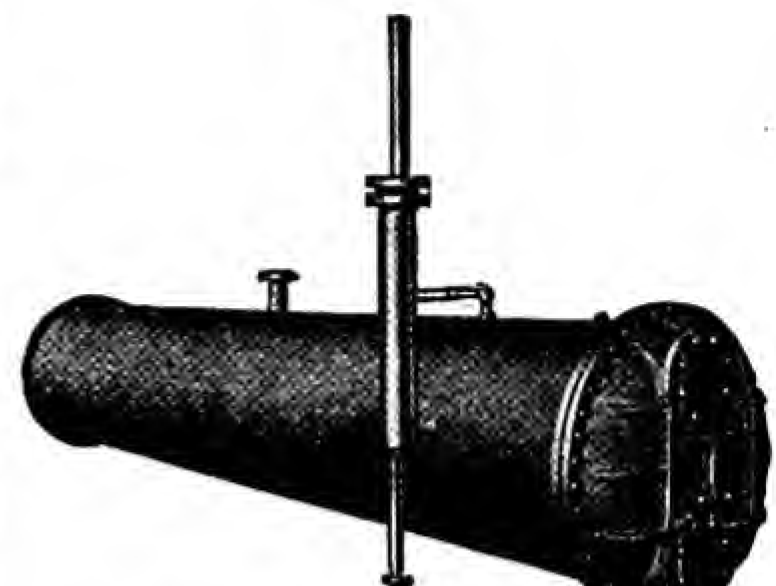


Fig. 113.—Shell-and-Tube Brine Cooler with Ammonia Level Indicator.

The double-pipe and shell-and-tube brine coolers may be protected somewhat from harm due to freezing of the brine in the tubes by installing an alarm on the brine discharge line from the cooler to give a suitable warning when the flow of brine is stopped or partially stopped, due to the clogging of the tubes with ice.

Shell-and-tube brine coolers may be installed in vertical position also, thus allowing the brine to drop through the tubes in one single pass. In this case the top of the shell is provided with a brine distributing box; the base is provided with a brine collecting tank; the cooler then operates on the open system.

Brine coolers are usually insulated when installed in insulated rooms. Any of the foregoing forms of brine coolers may also be used for the cooling of water. Of course if double-pipe or shell-and-tube types are used for cooling water, then special precautions must be taken to prevent freezing of water in the pipes or tubes.

The following tabulation gives the amount of shell-and-tube brine cooler surface in square feet required for one ton of refrigeration and various temperatures of brine and refrigerant. The tabulation is based on a heat transfer coefficient of 80 Btu. per sq. ft. per deg. of mean temperature difference.

Table 65.—Shell-and-Tube Brine Cooler Surface Sq. Ft. Per Ton of Refrigeration.

Av. Brine Temp.	Refrigerant Temp. Deg. F.									
Deg. F.	–25	–20	–15	–10	–5	0	5	10	15	20
25					5.0	6.0	7.5	10	15	30
20				5.0	6.0	7.5	10	15	30	
15			5.0	6.0	7.5	10	15	30		
10		5.0	6.0	7.5	10	15	30			
5	5.0	6.0	7.5	10	15	30				
0	6.0	7.5	10	15	30					
–5	7.5	10	15	30						
–10	10	15	30							
–15	15	30								
–20	30									

Brine Circulation Systems.—Cold brine may be distributed in the refrigerated room by either of two methods. In the first method, the brine pump takes the warm brine from a collecting tank in the machine room and discharges the brine through the brine cooler into the piping

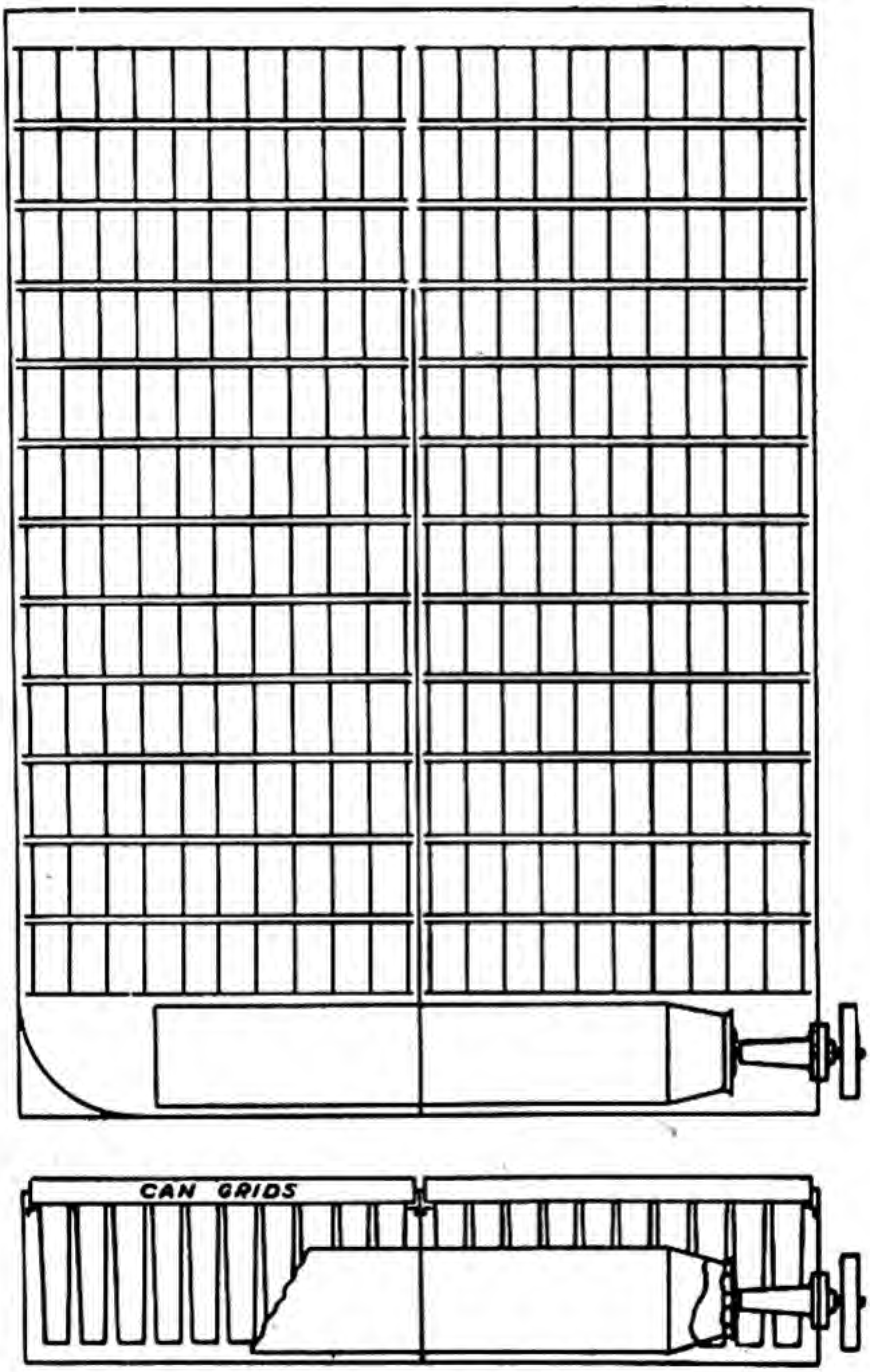


Fig. 114.—Typical Arrangement of Tank Using Brine Cooler.

system. After passing through the brine coils, it returns again to the machine room. In order to keep the pipe full of brine and to keep the brine under pressure, a reducing valve is placed on the brine return

line at a point just before it enters the collecting tank. This type of circulation system may be used when the height to which the brine must be pumped is not excessive.

In larger installations and in plants where the head against which the brine must be pumped is large, use is made of the balanced system of circulation. This system is shown diagrammatically by Fig. 115. This figure shows a closed brine system as applied in a modern packing house. A shell-and-and tube brine cooler is shown. A balance tank is located on top of the building, well above the highest brine coil. The brine pump and coolers may be located in the basement or on the first floor. The pump suction is taken from the balance tank and the pump discharges the warm brine through the coolers.

The cold brine leaving the brine coolers passes through the coils in the various rooms and the brine is then collected and returned to the balance tank at the top of the building. The pumping head is thus reduced to the difference of the levels of the brine in the tank and the top of the tank. The power required for pumping is that for overcoming the pipe friction, producing the velocity of the brine, and for overcoming the other small losses. The pipes are maintained full of brine in a positive manner at all times in this system.

A defrosting system is provided for the coils in the hog and beef coolers. This is accomplished by keeping the coil surfaces wetted with brine. As moisture is absorbed by this brine, a concentrating apparatus must be provided as shown. The concentrator is simply an apparatus for evaporating some of the water out of the brine by the application of heat.

Fig. 116 illustrates a balanced brine system, which uses the spray cooling system in the beef and hog coolers.

Amount and Kind of Brine.—Calcium chloride brine is generally used, since it has desirable operating characteristics. It does not seem to attack the metals of the system so readily as brine made from sodium chloride and it has lower freezing temperatures. In packing houses and other industrial applications where the brine is distributed in an open flow through nozzles and the like, sodium chloride brine is generally used. This is due to the fact that occasional splashing of sodium chloride brine on commodities does not have an injurious effect.

After the kind of brine has been determined, the amount to be circulated may be readily estimated. Under general considerations, the temperature range of the brine will be found to vary from 3° F. to 6° F. in passing through the room coils. The temperature range and the specific heat directly determine the amount of brine to be circulated to absorb a given quantity of heat. The heat absorbed by the brine may be determined by the following formula:

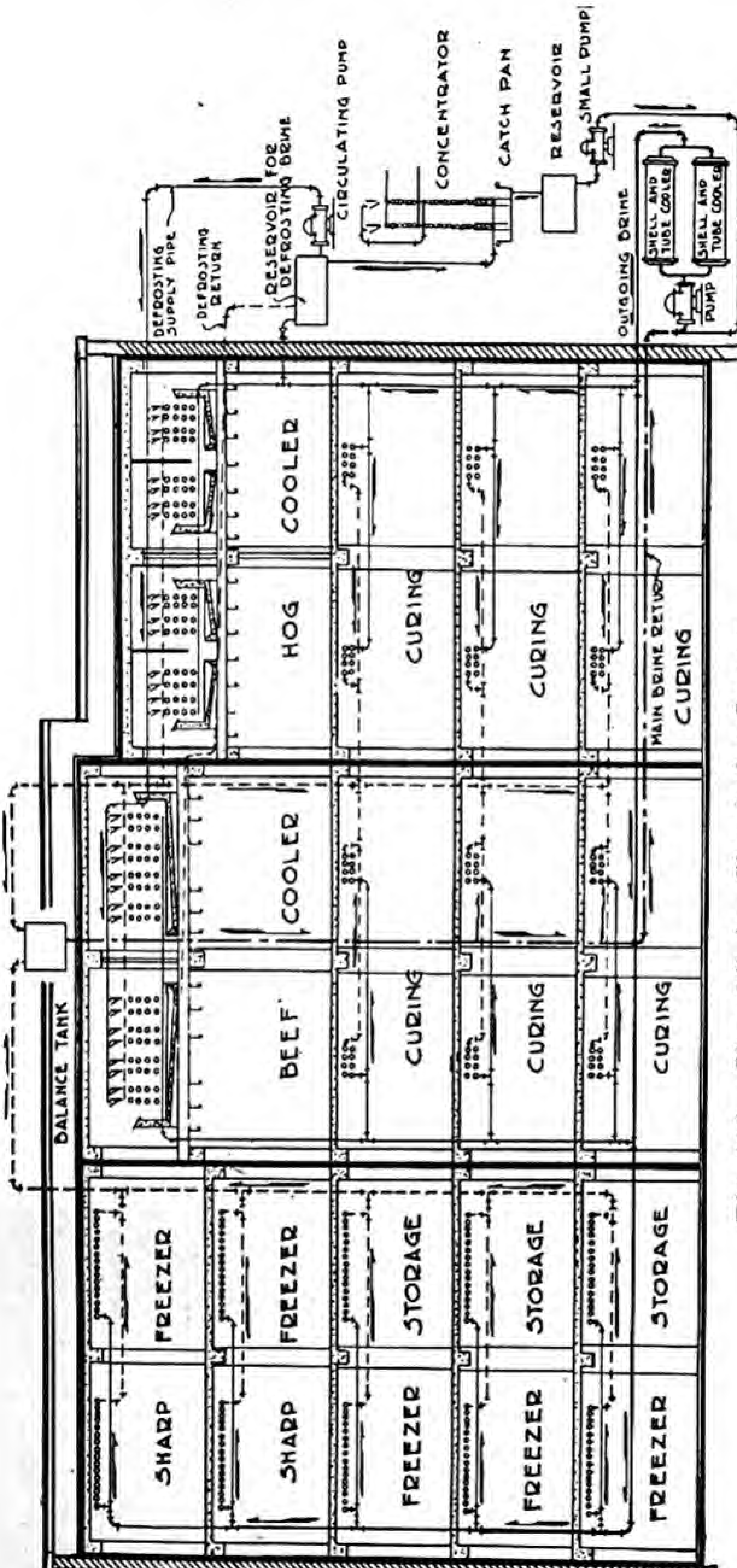


Fig. 115.—Closed Brine Circulating System.

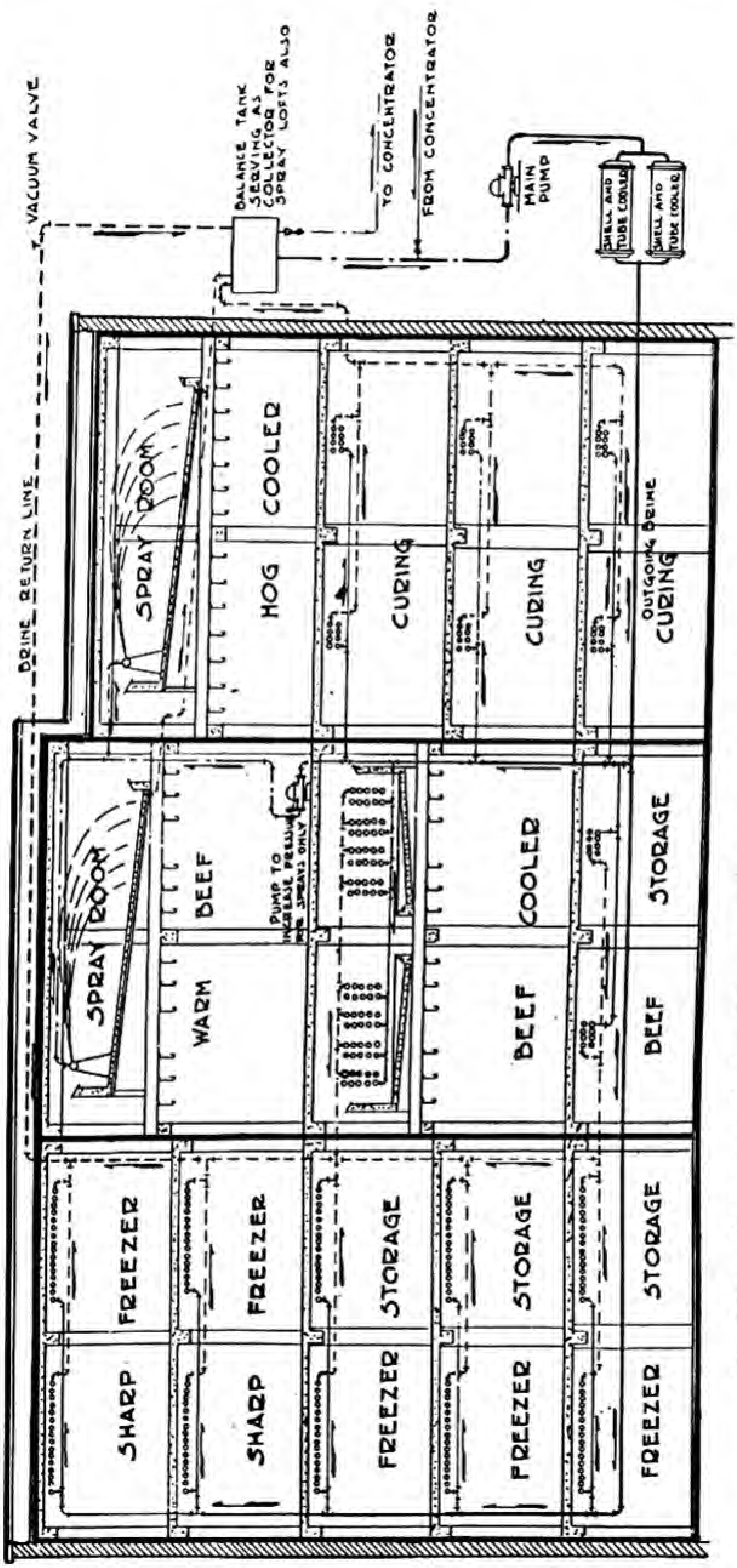


Fig. 116.—Balanced Brine Circulating System with Brine Spray Lofts.

From "Modern Packing House," Nickerson & Collins Co.

$$H = W \times S \times (t_1 - t_2)$$

where H = heat to be absorbed
W = weight of brine in pounds
S = spec. heat of brine
t_1 = higher brine temperature
t_2 = lower brine temperature

Thus, it may be desired to determine the amount to be circulated per min. per ton of refrigeration, when calcium chloride brine, having a specific gravity of 1.200 and specific heat of 0.700 and warming from 8° F. to 12° F., is used.

$$200 = W \times 0.700 \times (12^\circ - 8^\circ)$$

$$W = \frac{200}{0.700 \times (12 - 8)} = 71.4 \text{ lbs. per min.}$$

Since the specific gravity is 1.200, a gallon of the brine would weigh $8.33 \times 1.200 = 10.0$ lbs. approximately. The gals. per min. per ton of refrigeration would be found as follows:

$$71.4 \div 10 = 7.14 \text{ g.p.m. per ton}$$

Forced Air Circulation System.—The forced air circulation system for distributing refrigeration is sometimes termed the indirect system of refrigeration. In this system, air is cooled and then blown into the rooms or spaces to be refrigerated. The refrigerating coils, which contain brine or volatile refrigerants, are placed generally in a space called the bunker room. A fan draws in the air from the outside or from the rooms, and then discharges the warm air across the refrigerating coils in the bunker room, which is a sort of central cooling station for the several rooms of the plant.

After being cooled, the air leaves the bunker room and passes into a duct system which is connected to each room or space to be so refrigerated. The air, upon being circulated through the rooms, rises a few degrees in temperature, thereby producing the desired refrigerating effect in the rooms. The air thus produces the refrigeration by means of its sensible heat. In most cases, the warm air is collected by a duct system and is then led back to the suction side of the fan to be mixed with fresh air from the outside.

Treatment of the Air.—Practically, certain refinements must be introduced into the system in order that desirable results may be obtained. The air from the outside often contains excessive amounts of moisture, dust, smoke, etc., which would make it unfit to be introduced into a room in a cold storage warehouse. It is therefore necessary to wash the air before it is used. The fan generally draws in the fresh

air from the outside through a shower of water and then discharges it through the bunker room. Moisture will freeze on the refrigerating coils, so that suitable means of keeping the coils free from ice and frost must be adopted.

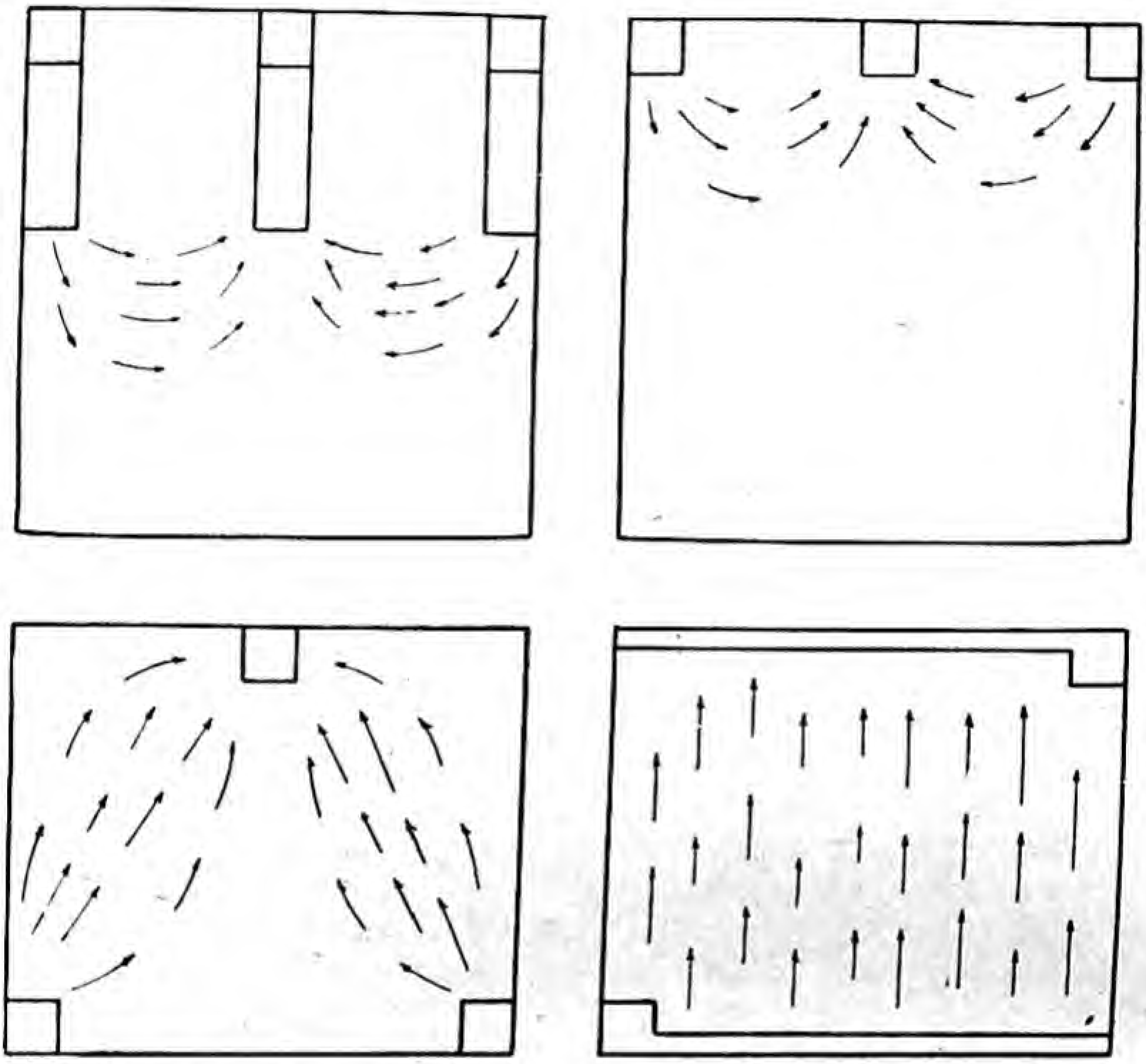


Fig. 117.—Forced Air Circulation Distribution Systems.

As the air leaves the bunker room it is, of course, in a saturated condition, so that the humidity must be regulated in some manner. This is usually accomplished by cooling the air several degrees below the room temperature and then by heating it or by exposing the air to some drying substance such as calcium chloride in the solid form.

Location of Air Ducts in Rooms.—The proper location of the cold and warm air ducts in the cold storage rooms and spaces is a consideration of primary importance to the successful operation of the system. The duct system should be arranged so that there is a uniform and vigorous circulation in all parts of the rooms. Various methods of distributing the air have been devised. These consist of ducts located

on walls, floors, and ceilings, in connection with perforated walls, ceilings and floors. Various arrangements are shown by Fig. 117. Warm and cold air ducts, located on the ceiling, are used quite extensively. The arrangement with cold air duct and perforated floor, and warm air duct and perforated ceiling gives a uniform air circulation in all parts of the room. It will be observed that the local conditions in the plant have quite a bearing on the design of the air distribution system.

Amount of Air Required.—The amount of air to be circulated depends upon the quantity of heat to be absorbed and the permissible rise of temperature. Generally, when the fan draws air from the room directly, the temperature rise will vary from 3° to 8° F., which will produce desirable conditions in the room.

Thus, the volume of air to be circulated to absorb heat at the rate of one ton of refrigeration per day when warming from 32° F. to 37° F. may be determined as in the following: From the formula given in Chapter XVI, the approximate specific heat of the air at the room temperature may be determined to be 0.2414.

The weight of air in lbs. per min. may be found from the following formula:

$$H = W \times S \times (t_1 - t_2)$$

where H = heat to be removed
W = weight of air in pounds
S = specific heat of air
t_1 = outlet air temperature
t_2 = inlet air temperature

Then, since $S = 0.2414$, $t_1 = 37$, and $t_2 = 32$, the weight is readily found:

$$200 = W \times 0.2414 \times (35^\circ - 32^\circ)$$

$$W = \frac{200}{0.241 \times 5} = 165 \text{ lbs. per min}$$

Also, from Table 90, page 495, the volume of air per lb. will be seen to be approximately 12.5 cu. ft. per lb. The volume of air per min. per ton is then equal to:

$$165 \times 12.5 = 2060 \text{ cu. ft.}$$

After the various volumes of air to be used have been ascertained, the duct systems may be laid out, the size of the fan and motor determined, and all other details of the system completed.

Considerations of Forced Air Circulation.—The following are some of the most important considerations of the forced air circulation for distributing refrigeration. By means of this system it is possible to maintain an accurate control over the temperatures in all parts of the

rooms, the relative humidity, and the purity of the air. By maintaining the correct humidity, excessive evaporation of the moisture from the materials stored in the rooms may be avoided. By proper arrangement of the duct system, not only will uniform temperatures be maintained in all parts of the room, but also much more space is available for storing materials. The relative amount of pipe surface for cooling the air is appreciably smaller than when the gravity air circulation system is used. With a properly designed duct system and a suitable fan, the power for circulating the air should amount to only a small quantity. Due to the fact that the system as a whole has several desirable characteristics, it is used and should be used in the larger cold storage rooms.

Unit Type Air Coolers and Conditioners.—One of the latest developments in the distribution of refrigeration in cooled rooms and buildings is the unit type air cooler and conditioner.

The use of air cooling and conditioning equipment of the forced air type has been quite extensive in many commercial applications of refrigeration, such as the cooling of theatres and auditoriums, and the cooling and conditioning of air to produce definite temperatures, as well as certain humidities in industrial plants where such requirements are necessary. In most of the earlier applications of this kind the air cooling and conditioning machines were of the large centralized type.

One of the latest developments along the same lines as used in the larger plants is the design and perfection of small unit type air cooling and conditioning machines which may be used for producing refrigeration in the smaller rooms. These unit type air coolers and conditioners are constructed as a complete air cooling and conditioning machine with all the essential products incorporated in same.

The extensive use of unit type air coolers and conditioners for small and medium size rooms is due to a number of factors. In the first place when unit type air coolers and conditioners are used for refrigerating rooms it has been found that better conditions are produced in the room. In other words, goods which are stored in rooms cooled by this type of equipment keep better due to the fact that there is a positive circulation of the air throughout the room, there is a uniform temperature throughout the room and the goods, thus eliminating dead air pockets and stagnant air which tends to increase the growth of mold and other deteriorating effects on the goods.

The use of the forced air principle in distributing the refrigeration in the room also tends to decrease shrinkage of the products stored in the room. Generally it will be noticed that the air is purer in a room

cooled by unit air coolers and conditioners than when the conventional type of refrigerating coils are used.

Mechanically speaking, the units are self-contained, are compact, and occupy a minimum amount of space in a room. They also eliminate expenses due to construction of overhead drip pans and bunkers which are sometimes necessary when refrigerating piping is used in the room.

Another advantage which sometimes is quite important is the facility with which the coils may be defrosted. In the case of using dry cooling coils in the unit air cooler, the frost will, of course, tend to collect on these coils, but in this case it is a much simpler matter to defrost a small bank of coils in a unit cooler than to defrost several hundred feet of piping distributed about the room and on the ceiling of a room.

These things, in combination with others, are the reasons why unit type air coolers and conditioners have been developed, perfected, and are now in extensive use for producing refrigeration in all kinds and sizes of rooms at various temperatures, and they will continue to be used more and more.

In the conventional design of unit type air cooler or conditioner, the air to be cooled is usually drawn in at the bottom of the casing, passes upwardly through the refrigerating coils and is then discharged by suitable fans through air distributing elbows or duct work into the room to be cooled. The air circulates through the room and then returns to the bottom of the casing. The refrigerating coils are located inside of the casings directly in the path of the air.

Unit type air coolers may be divided into two principal classes. In the first class are those which contain dry refrigerating coils.

In the case of the use of a volatile refrigerant, these coils may be constructed for the refrigerant in question, such as ammonia, carbon dioxide, sulphur dioxide, methyl chloride, etc.

Also, if it is desirable, brine cooled in a central cooling plant may be circulated through the coils located in the air coolers. In this class of air coolers the problem of defrosting the coils may be handled in one or two ways. If the temperature of the refrigerated room is somewhat about 32° F., it is possible to install a number of unit air coolers so that each may be periodically shut down to allow the coils to defrost.

Another method of defrosting the coils in this type of air cooler consists of a brine tank located under the unit, a brine pump for pumping the brine from the brine tank into suitable spray nozzles located above the pipe coils for the purpose of periodically spraying brine over such coils to remove any accumulation of ice or frost.

In the second class of unit type air coolers and conditioners are

those which are equipped with tanks, pumps and spray nozzles for continually spraying water or brine over the cooling coils, as illustrated by Fig. 118.

In this type of air cooler the air is drawn in at the bottom of the casing and passes upwardly through the refrigerating coil and brine or water spray, then through suitable eliminating plates into the fans which discharge the air into the room. The air in passing through this type of unit is cooled principally by the action of the spray and the brine or water in turn is cooled by its contact with the refrigerating coil.

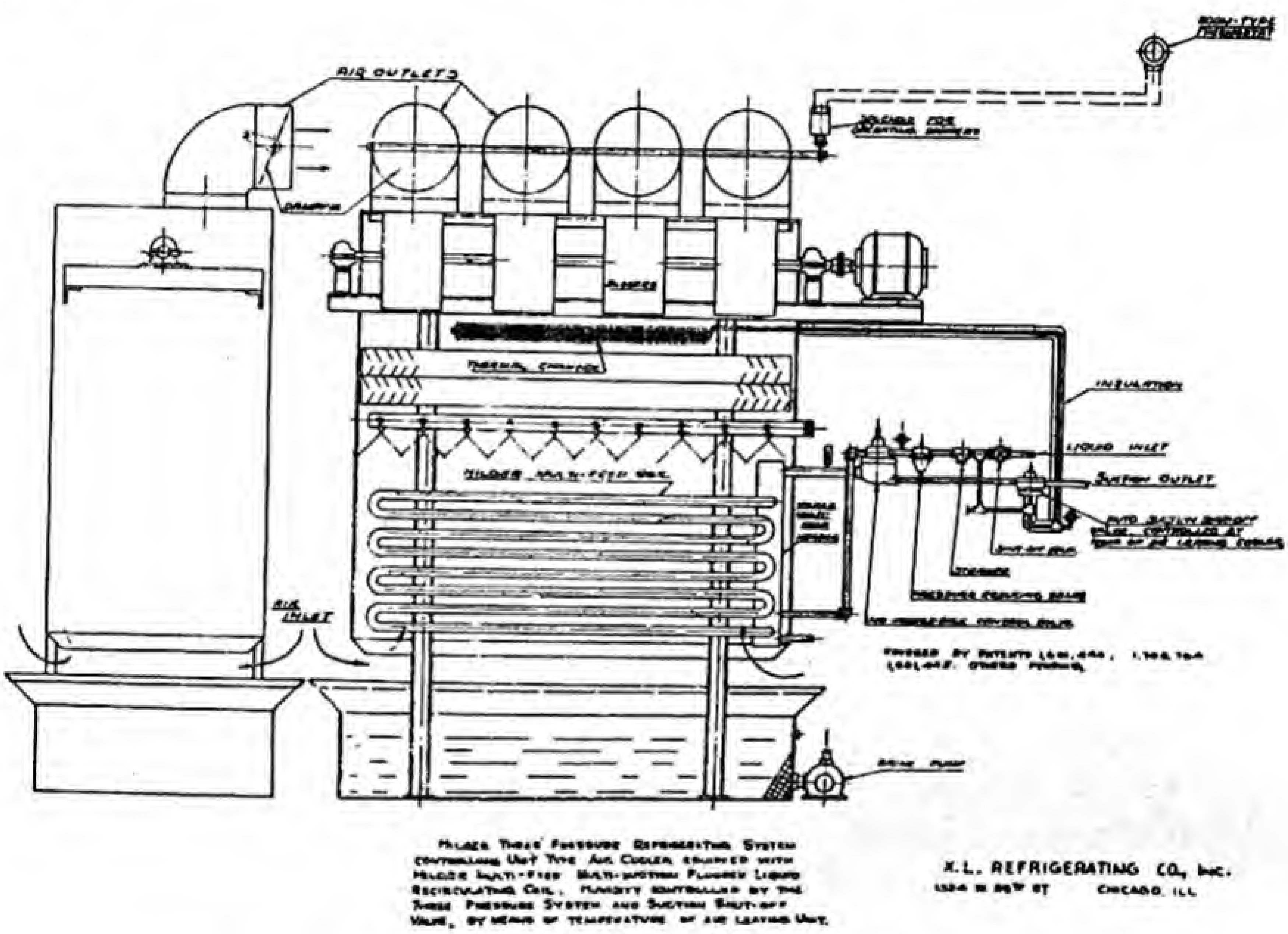


Fig. 118.—Hilger Unit Type Air Cooler and Conditioner.

This type of unit may also be equipped with refrigerating coils suitable for any of the well known refrigerants. They also may be equipped with brine or water coils for receiving brine or water which has been cooled in some central cooling plant.

In the case of the use of an air cooler or conditioner equipped with sprays for continually spraying brine or water over the cooling coils, the problem of defrosting the coils is practically eliminated. In the case of the use of brine, it is only necessary to re-strengthen the same at intervals to compensate for the moisture which has been condensed out of the air.

A number of various materials of construction have been used in the fabrication of air coolers and conditioners. The drip-pans, tanks, and casings located under these units are sometimes made of copper bearing steel, galvanized and painted. The refrigerating coils, either brine or ammonia type, are always galvanized.

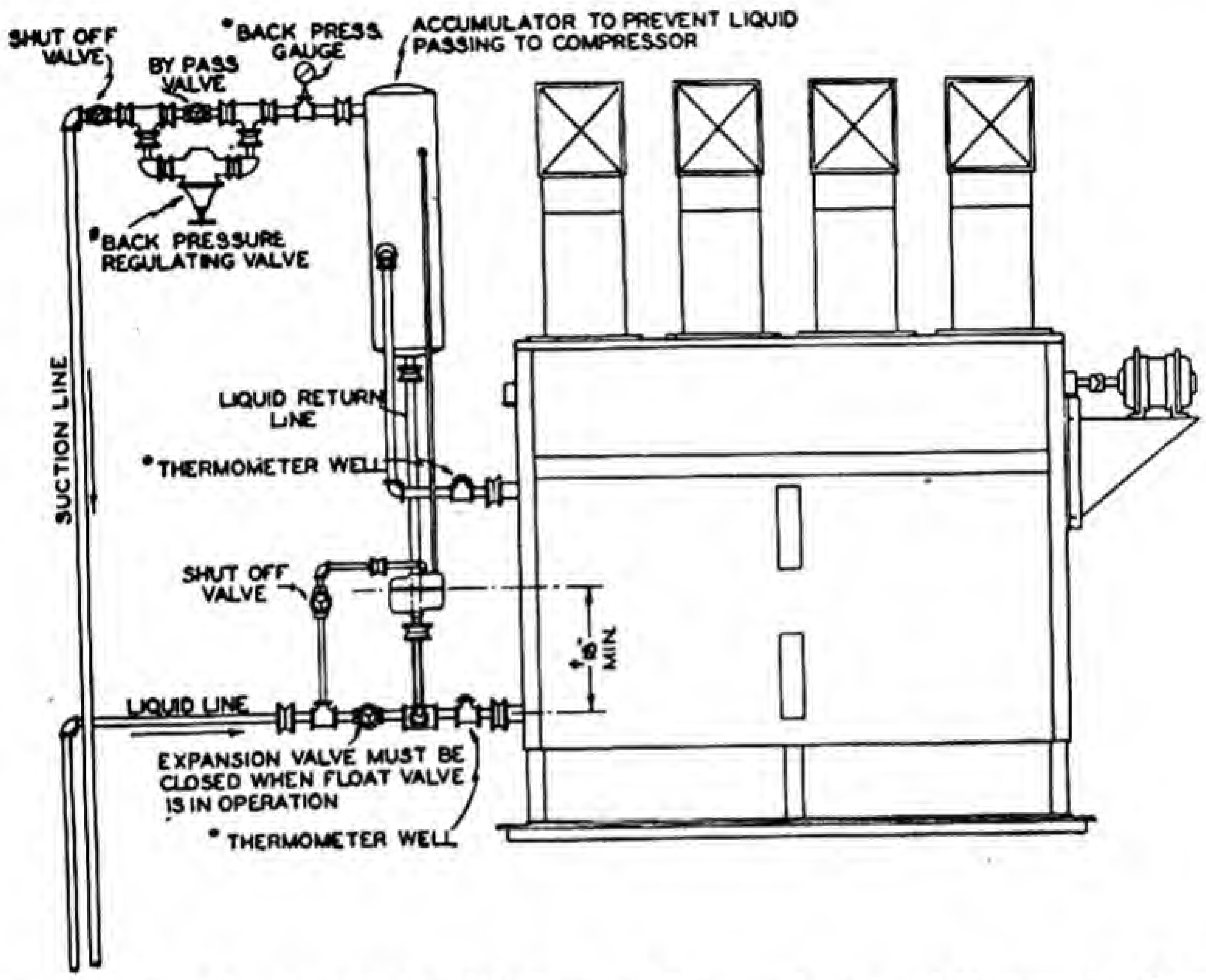


Fig. 119.—Air Cooling Unit with Suction Accumulator and Automatic Control.

In the square type unit air coolers and conditioners use is commonly made of a number of multi-blade fans mounted on a rigid fan shaft. This fan shaft is usually direct-connected to a motor by means of a flexible coupling. Both motor and fan shafts are supported on either ball or roller bearings.

The flow of refrigerant to the refrigerating coils in the unit may be under either hand or automatic control. In most cases in which ammonia is used as a refrigerant, these units are commonly controlled by a device which automatically feeds the correct amount of ammonia to the coils in proportion to the actual refrigerating load on the same.

In the case of the use of brine for the refrigerating medium, this may also be accomplished by means of an automatic brine control valve.

In addition to the control of the refrigerant to the evaporating coils, additional control may be added to the unit for the purpose of automatically maintaining a given temperature in a given room.

The spray type air cooler and conditioner may be equipped with humidity controls for maintaining automatically the humidity desired in any room at any temperature.

Both the dry and wet coil types of coolers and conditioners may be equipped with heating coils so that rooms in which these units are installed may be heated, should this be necessary.

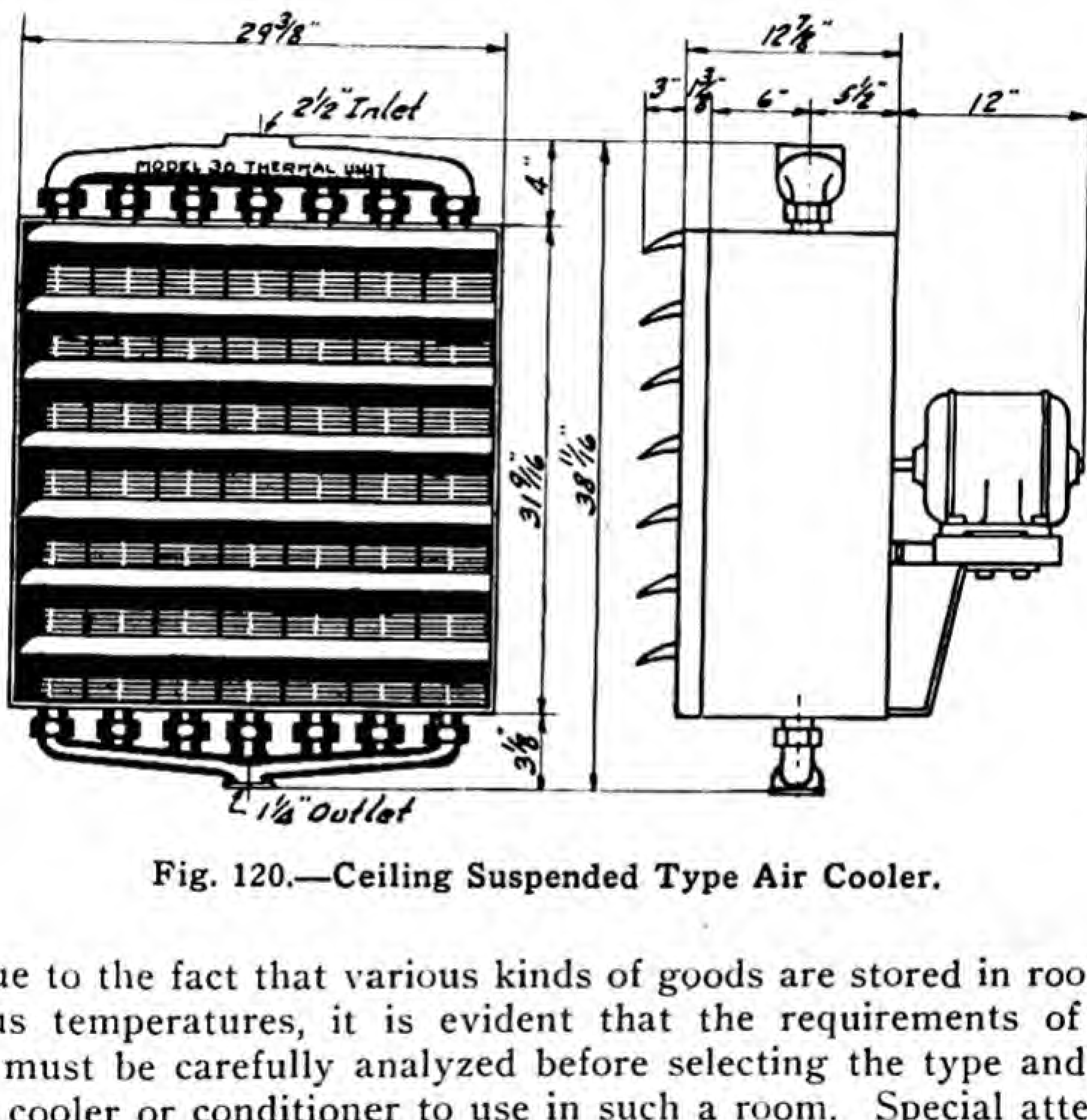


Fig. 120.—Ceiling Suspended Type Air Cooler.

Due to the fact that various kinds of goods are stored in rooms at various temperatures, it is evident that the requirements of each room must be carefully analyzed before selecting the type and kind of air cooler or conditioner to use in such a room. Special attention must be given to kind of goods stored, the amount of the goods stored, temperature desired, humidity to be maintained, and the distribution of the cooled and conditioned air.

Unit type air coolers and conditioners have been found to be especially useful for producing refrigeration in cold rooms, such as meat cooling and storage rooms, sausage cooling and manufacturing rooms, fruit and vegetable precooling and holding rooms, egg and butter storage rooms, milk and cream storage rooms, cold rooms in confectioneries and bakeries, etc.

Unit type air coolers and conditioners are usually manufactured in all sizes and styles suitable for the smallest to the largest of rooms. In larger rooms more than one machine may be used.

Unit type air coolers and conditioners have passed entirely through the experimental stage, and at present really constitute an important factor in the distribution of refrigeration in cooled rooms and buildings.

Fig. 118 illustrates the construction of the Hilger unit type air cooler and conditioner as made by the X L Refrigerating Co. This unit is of the spray type. It is equipped with the necessary eliminators, brine pump, and automatic ammonia controls.

Fig. 119 illustrates the piping connections used on the floor type air cooling unit. It is also equipped with automatic ammonia controls.

Fig. 120 illustrates the construction of small cooling units of the ceiling suspended type. The face of this unit is equipped with air-directing louvers. The air is forced through the unit by means of a direct-connected motor-driven propeller fan.

Extended Surfaces for Evaporating Units.—Various forms of extended surfaces have been used recently on small evaporating units for small refrigerators, compartments, and coolers. These extended surfaces are usually flat or curved surfaces or plates which are attached to the primary evaporating pipe. Various materials and arrangements are used. Due to the particular disposition of the pipes and extended surfaces, these evaporating units are commonly termed "fin coils."

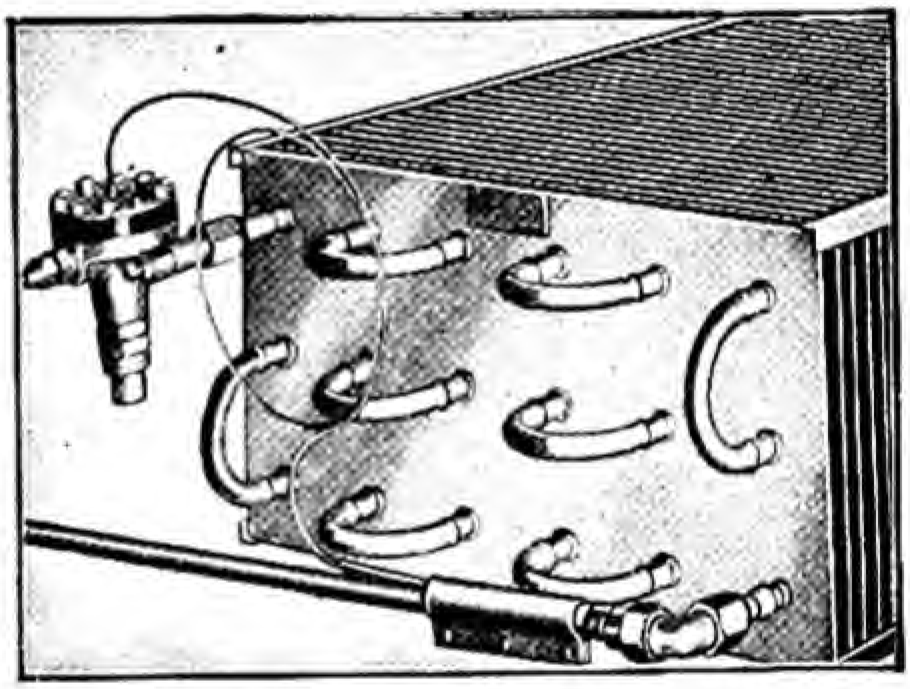


Fig. 121.—Larkin Coil Showing Details of Construction.

A Larkin fin coil evaporating unit is illustrated in Fig. 121. In this coil the plates or fins are made of aluminum. The tubes or pipes are made of copper. In fabricating the units especial attention is given to obtaining a good metallic joint between the fins and tubes.

Another type of extended surface coil is illustrated by the Hilger "spiral-fin" coil, as shown in Fig. 122. In this coil, use is made of steel pipe for making the coil proper and its multi-feed header. The fin surface consists of helical shaped ribbons of steel which are attached to the pipes. The whole unit is hot-dipped galvanized after fabrication; thus insuring good contact between the fin surface and the pipes.

The relative efficiency of the extended surfaces depends upon a number of factors such as kind of metal used, spacing of tubes or pipes, spacing of fins, method of attaching fins to pipes, etc.

Generally speaking, the total surface of a fin coil will be considerably more than that of a plain coil used for the same purpose. Due to the use of more surface, its temperature may be carried at a higher point, thus maintaining a higher humidity in the refrigerated compartment. Fin coils are made in various sizes and shapes to suit the requirements of any refrigerator or cooler.

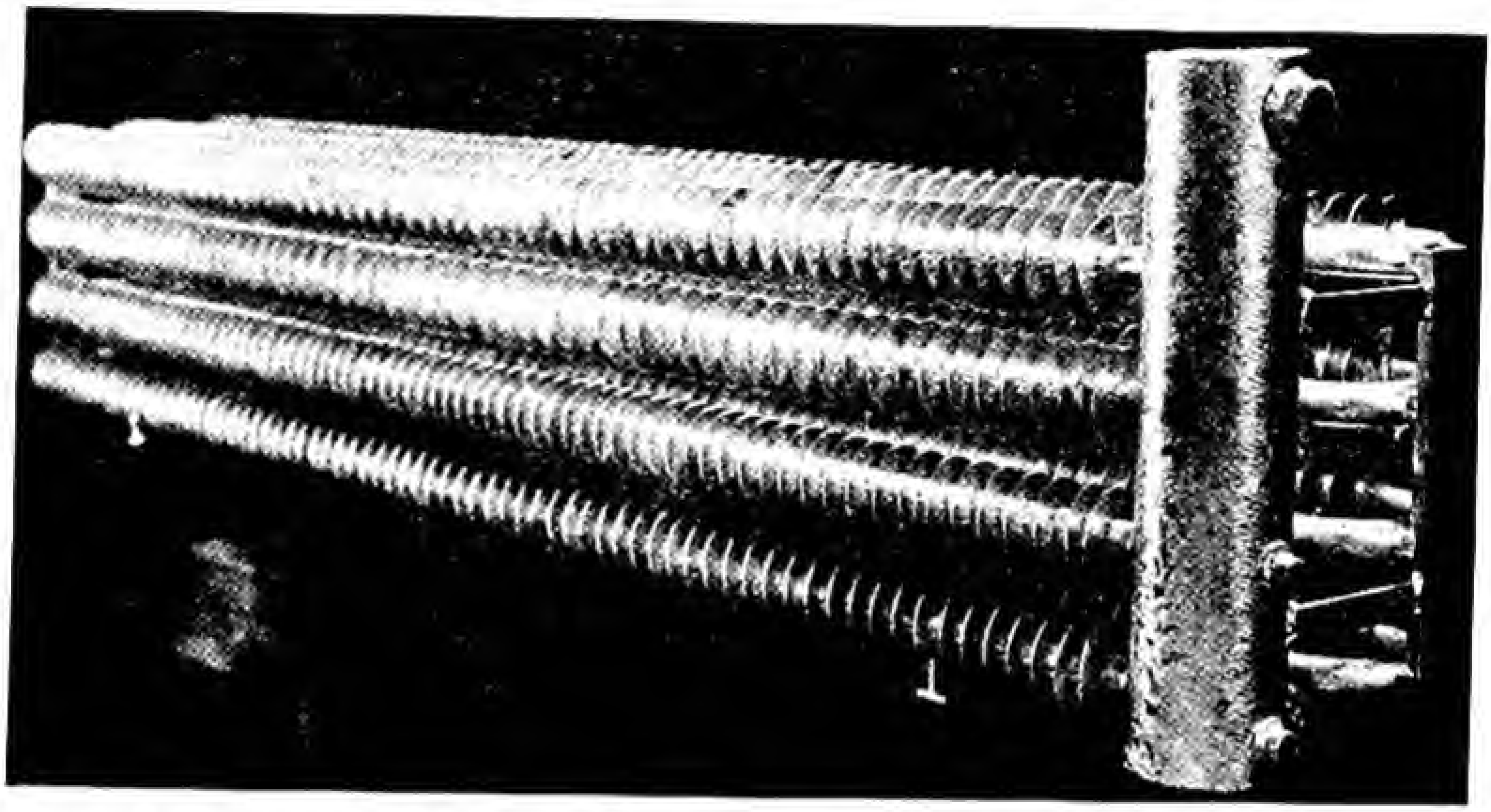


Fig. 122.—Hilger Spiral Fin Coil and Multi-Feed Header.

QUESTIONS ON CHAPTER IX.

1. Explain what is meant by "Direct-Expansion" of refrigerants.

2. Explain how the location of coils and the use of aprons and baffles for refrigerating coils affect the circulation of air in the cold storage rooms.

3. How many feet of 2-in. direct-expansion pipe should be installed in a cold storage room having a temperature of 40° F. if 576,000 Btu. must be removed every 24 hours, when the temperature of the ammonia is 15° F.?

4. Explain the general principles of the action of holdover and congealing tanks.

5. Describe the principles underlying the operation of the flooded system of evaporation, and enumerate its characteristics.

6. Name some of the advantages of the brine system for distribution of refrigeration.

7. Describe the balanced brine circulating system and enumerate its advantages.

8. How many gallons of calcium chloride brine must be circulated per minute to absorb heat at the rate of 100 tons per day of 24 hours if the brine warms from −12° F. to −8° F. in passing through the cold storage rooms?

9. Describe the forced-air circulation system and discuss its characteristics.

10. How many Btu. will be removed from the cold storage room by 15,000 cu. ft. of air, warming from 32° F. to 35° F.

CHAPTER X.

AUTOMATIC REFRIGERATION SYSTEMS.

General Consideration.—The demand for automatic control of refrigerating plants has depended upon the development of the smaller refrigerating installations. During recent years many small refrigerating systems have been installed in meat markets, restaurants, apartment houses, hospitals, etc. In the majority of cases, these plants have been electrically driven, a condition which lends itself readily to the use of automatic control. Another consideration in favor of automatic control in these small refrigerating systems is the lack of proper attendance. Generally, the operator of these small systems has many other duties in addition to the operation of the refrigerating system. Furthermore, in most cases it is necessary only to operate the refrigerating system for a part of the day. Thus, the systems are generally shut down during the nights.

One of the earlier methods of producing a uniformity of temperature in these small refrigerated rooms was a system of holdover and congealing tanks. These contained cold or solidified brine which supplied refrigeration during the long shutdown periods of the compressor. While these systems maintained a fairly uniform temperature, additional space had to be provided for the tank. In consideration of these conditions, many of the smaller modern refrigerating systems are operating on the automatic basis.

In general, the system consists of, in addition to the regular refrigerating systems, a thermostat and an automatic electric control for stopping and starting the compressor motor, according to the variation of temperature in the cold room, together with an automatic expansion valve.

Automatic Expansion Valve.—In order to maintain a constant suction pressure on the refrigerating system it is necessary to use an automatic expansion valve of which there are many different types on the market. There are several of the common types of construction. In one type the admission of the liquid to the low pressure side of the system is controlled by a spring operating against a steel plate

diaphragm. In this kind of valve it will be observed that when the pressure on the liquid outlet side of the valve is reduced below a certain pre-determined point, the spring will act against the stem of the valve, thereby producing a slight opening at the seat. The liquid refrigerant, by flowing through this small orifice, reduces its pressure from that of the high pressure side to that of the low pressure side. The valve really operates on the well-known principle of a reducing valve.

It is provided with a suitable strainer, through which the liquid refrigerant passes before it reaches the valve seat. Means are provided for adjusting the action of the spring against the steel plate diaphragm. This controls the pressure below which the spring will cause the valve to open. Suitable means are provided for removing the working parts of the valve for the purpose of regrinding the valve seat, should this become necessary.

Thermostats.—For controlling the operation of an electric motor driving the compressor, a thermostat is installed in the cold room. These thermostats are so constructed that when the temperature rises to a certain point, an electric contact is made thereby starting the compressor motor. After the temperature has been reduced to a certain pre-determined point, the thermostat makes another electrical contact which stops the compressor motor. These thermostats depend upon expansion and contraction of matter with the change of temperature for the making of the electrical contact. They are so constructed that they are very sensitive to changes of temperature, and so that they may be adjusted for different temperatures. Thus, the thermostat may be so adjusted that when the temperature rises to 44° F., it will make an electrical contact which starts the electric motor driving the compressor, and when the temperature is reduced to 40° F., another electrical contact is made which stops the compressor.

Excess Pressure Relief Valve.—In order to protect the refrigerating system in case of failure of condenser water supply or the incorrect manipulation of the valves on the system, it is generally advisable to provide valves for relieving the excess pressure. One of common type consists of a spring which gives movement to a plunger in proportion to the pressure. At a certain pressure, the plunger will be so displaced that it will operate a trigger which is connected to the main electrical switch or a suitable safety valve.

To protect the refrigerating system against damage when the pressure on the high pressure side is suddenly reduced to zero, a low pressure relief valve is sometimes installed. One type of construction is very similar in construction to the excess pressure type. When

the pressure on the high pressure side drops below normal, due to a rupture in the system somewhere, the plunger is so operated by a spring that the weighted lever is tripped. This weighted lever can be used to control either the quick closing valve, or the main electric switch.

Thermostatically Controlled Valves.—For the purpose of facilitating the operation of the automatic refrigerating systems, thermostatically controlled valves have been devised. These valves are used for the purpose of controlling the flow of the refrigerant, water or brine. One valve is driven by a motor which is controlled by the operation of the thermostat. The motion of the motor is transmitted by suitable gears and cams to the valve stem, the valve stem being held against the cam by means of a spring. The small motors for driving these valves are wired for either alternating or direct current at 110 or 220 volts. With these valves, it is possible to control the temperature in the individual rooms which are cooled by either brine circulation or ammonia expansion. This is accomplished by placing one of these motor driven valves on the brine or liquid line leading into the room, and by controlling this valve by thermostats located in the room.

Semi-Automatic Refrigeration Systems.—In some refrigeration installations it is desirable to have the system work on the semi-automatic basis. In some instances, the refrigeration may be needed after the regular working hours, or some other conditions may be present which will make it necessary for the operator to be absent from the plant when the machine is closed down. The machine in this case is started by hand, after which the thermostat circuit is closed. When the temperature has been reduced to the desired point, the thermostat acts to close down the system completely.

This system consists of an ammonia pressure circuit breaker, a pressure condenser water regulating valve, a thermostat, a thermostatically operated switch, and an automatic expansion valve. The system is started by placing a three-pole double-throw switch in the thermostat circuit in the down position, and by starting the compressor in the usual manner. The system is protected from higher pressure by means of the ammonia pressure circuit-breaker, and the flow of the water to the condenser is regulated by means of the pressure within the condenser.

When it is desired to have the thermostat shut down the system when the temperature has been lowered to a certain pre-determined point, the double-throw switch in the thermostat line is placed in the upper position. In this position the thermostat is connected to the system and will close down the compressor, as soon as the tempera-

ture has been lowered to the pre-determined point. After stopping the compressor, the automatic expansion valve immediately closes the liquid line, and as soon as the condenser pressure falls, the water regulating valve will stop the supply of water to the condenser.

Completely Automatic System.—In many installations, it is desired to have the refrigerating system start and stop itself; thereby performing its function in an automatic manner. A certain system ordinarily may be used when clear water at a pressure of 20 to 75 lbs. is available for absorbing the heat in the condenser. The system consists of a thermostatically operated motor-driven water valve, a water velocity actuated contactor for the compressor motor switch, an ammonia pressure circuit-breaker, a remote control motor switch, a thermostat, and an automatic expansion valve. The operation of this system may be described as follows: When the temperature rises to the predetermined point in the cold room, the thermostat makes the contact for the operation of the plant. This causes the small motor-driven valves to admit water to the ammonia condenser. The velocity of the water passing through the motor-driven water valves closes the circuit, which operates the remote control switch for the compressor. The compressor will be started in this manner, and will continue to operate until the thermostat makes a contact for stopping, until the condenser water fails, or until excess pressure is built up on the high pressure side.

The automatic expansion valve maintains the proper pressure in the cold room, corresponding to the temperature of the refrigerant to be maintained. Should bad water conditions exist, or should cooling tower water be used for the purpose of absorbing heat in the condenser, another type of system may be used.

This system is composed of a thermostatically operated motor-driven water valve for admitting water to the condenser, a rotary switch on the remote control circuit for the compressor, an ammonia pressure circuit-breaker, a thermostat, and an automatic expansion valve. In this system, when the thermostat makes the starting contact, the motor-driven valve admits water to the ammonia condenser. At the same time, the rotary switch closes the switch on the remote control circuit for the compressor. The system is started in this manner, and will remain in operation until the thermostat in the cold room makes a contact for stopping.

Temperatures and Pressures in Evaporators.—It is well to review at this time some of the fundamental considerations of temperatures and pressures to be maintained in evaporators, using various refrigerants for different refrigeration purposes.

In all cases of the refrigeration or cooling of substances, the temperature of the evaporating refrigerant must be maintained a few degrees below the temperature of the substances to be refrigerated, so that the heat will flow by natural tendency from such substances into the evaporating refrigerant. The evaporating temperature of refrigerant depends upon the pressure maintained upon such refrigerant during the evaporation process. Also, the relative temperature obtained by a given refrigerant, depends upon the refrigerant used, and is entirely an individual characteristic of the given refrigerant. In other words, for a given temperature of evaporation, the various refrigerants will have different evaporating pressures. Conversely, for a given evaporating temperature, the various refrigerants will have different evaporating or boiling pressures.

The pressures to be maintained upon some refrigerants, in order to obtain a boiling or evaporating temperature of 5° F., are illustrated by the following table:

TABLE 66.—PRESSURES OF REFRIGERANTS.

Refrigerant	Evaporating Temperature Deg. F.	Evaporating Pressure Lbs. Absolute
Ammonia (NH_3)	5°	34.27
Carbon Dioxide (CO_2)...........	5°	331.9
Sulphur Dioxide (SO_2)...........	5°	11.80
Methyl Chloride (CH_3Cl)........	5°	4.65
Freon 12 (CCl_2F_2)...............	5°	26.51

Table 66 illustrates the wide variation of pressures to be maintained on different refrigerants in order to obtain a given evaporating temperature.

Heat Removed by Evaporators.—The heat removed by the evaporators depends upon a number of factors such as:

1. Weight of refrigerant evaporated in a unit time.
2. Kind of refrigerant used.
3. Temperature of liquid refrigerant, before expansion or throttling.
4. Temperature of refrigerant leaving evaporator.
5. Quality or dryness of refrigerant leaving evaporator.
6. Quality of refrigerant before expansion valve.
7. Temperature of evaporation.

In Table 67 it is assumed that one pound of the refrigerant is evaporated in one minute. The effect of using different kinds of refrigerants is due principally to the fact that the various refrigerants have different and characteristic latent heats of evaporation. The temperature of the refrigerant may be either that corresponding to its pressure, or

it may be higher, in which case the refrigerant is superheated. In Table 67 it is assumed that the refrigerant leaves the evaporator at the saturated temperature corresponding to the pressure. It is evident that the refrigerant leaving the evaporator may contain un-evaporated particles of liquid, in which case it is called a wet mixture of liquid and vapor. In calculating the data for Table 67, it is assumed that the refrigerant leaves the evaporators without any particle of un-evaporated liquid, that is, it is dry and saturated vapor. Furthermore, it is evident that the liquid before the expansion valve may be at various temperatures, at or below the saturated temperature due to the condensing pressure. To simplify the data in Table 67, it will be assumed that the refrigerants have a temperature of 75° F. before they pass through the expansion or throttle valve. Also, it is possible that the refrigerant may contain un-condensed vapor, and non-condensable gases, at a point before the expansion valve. It is assumed that the refrigerant before expansion valve is all liquid and free from foreign substances. The heat contents for carbon dioxide are based on a reference temperature of 32° F., but all other heat contents in Table 67 are based on a reference temperature of −40° F.

TABLE 67.—HEAT ABSORBED BY VARIOUS REFRIGERANTS IN EVAPORATORS.

Refrigerant	Heat Content of Liquid at 75°	Heat Content of Liquid at 5°	Heat to Cool Liquid Btu.	Latent Heat Btu.	Heat Absorbed in Evaporation
Ammonia	126.2	48.3	77.9	565.0	487.1
Carbon Dioxide	30.96	−13.73	44.69	114.70	70.01
Sulphur Dioxide	38.32	14.1	24.22	169.4	145.18
Methyl Chloride	42.7	16.3	26.4	179.0	152.6
Freon 12	25.1	9.3	15.8	69.47	53.67

Table 67 has been based on the foregoing limitations and shows how the heats absorbed by various refrigerants in the evaporator differ, and are characteristic to each refrigerant.

An inspection of the table of properties of refrigerants shows that the latent heat of evaporation of a given refrigerant varies considerably with the temperature of evaporation. In Table 67, a uniform evaporating temperature of 5° F. for all refrigerants will be assumed.

The net amount of heat absorbed by a refrigerant in the evaporator is found by subtracting the heat required to cool the liquid to the temperature of the evaporator, from the latent heat of the evaporation.

Table 67 shows that the amounts of heats absorbed by the various refrigerants vary considerably with the individual kind used. The heats absorbed in the evaporator are, of course, the net available refrigerating effects for the conditions previously stated, expressed in Btu. per lb.

The foregoing considerations in regard to temperatures, pressures, heats, etc., are essential for the intelligent construction and operation of devices and apparatus for the distribution of refrigeration.

Expansion Valves.—The primary function of the so-called expansion valve is to reduce the pressure of refrigerant from that existing in the condenser to that existing in the evaporator, and also is used to regulate the quantity of ammonia required in the evaporator. The word "expansion" does not describe its function correctly. Probably, it could be called a throttle, or pressure reducing device, more correctly. So long as there is a difference of temperature between the evaporator and condenser it is evident that there must be a difference of pressure also. The expansion valve must therefore produce the drop of pressure, and the drop of pressure is equal to the increase of pressure produced by the compressor.

Expansion valves may be divided into two general classifications, namely, manually operated, and automatically operated valves. The hand or manually operated expansion valves are usually of the needle type, and require no further description here. The automatic expansion valves are constructed along various lines, as indicated by the descriptions in the following paragraphs. Such automatic expansion valves and liquid flow control devices are actuated by various means, such as suction pressure, condenser pressure, evaporation temperature, condenser temperature, or combination of these things.

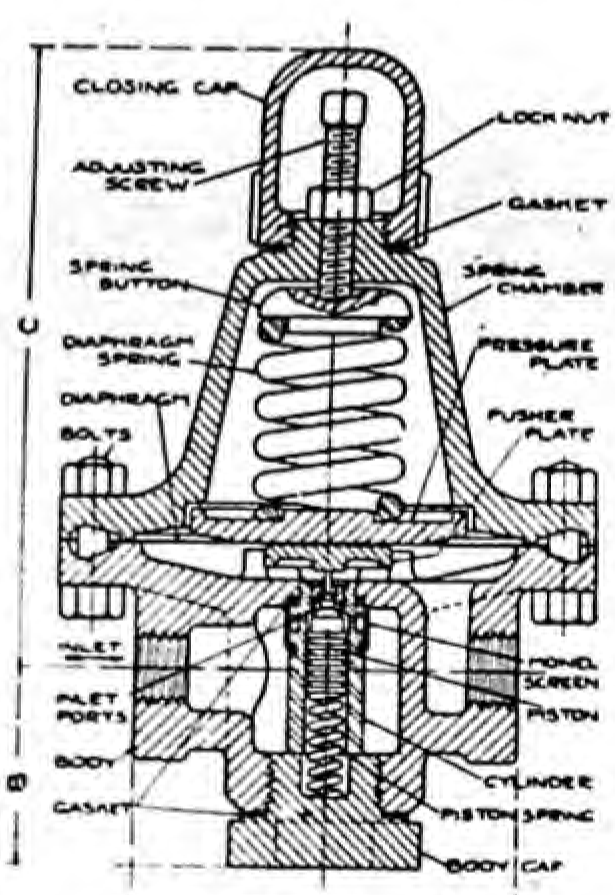


Fig. 123.—Cash Automatic Expansion Valve.

The construction of an automatic ammonia expansion valve of the diaphragm type is illustrated in Fig. 123. This type of valve may be set to maintain a constant suction pressure regardless of the fluctuations of the condenser pressure.

In Fig. 124, "2" shows the relation of the valve stem and the seat when the valve is closed; "3" and "4" show the relations of the stem and seat when valve is partially open and fully open.

TABLE 68.—DIMENSIONS OF AUTOMATIC EXPANSION VALVE.

(Letters Refer to Fig. 123.)

Size	1/4-in.	3/8-in.	1/2-in.	3/4-in.
A	4 1/4	4 1/4	4 5/8	4 3/4
B	2 1/2	2 1/2	2 5/16	2 1/4
C	6 1/2	6 1/2	7 1/4	8 1/4

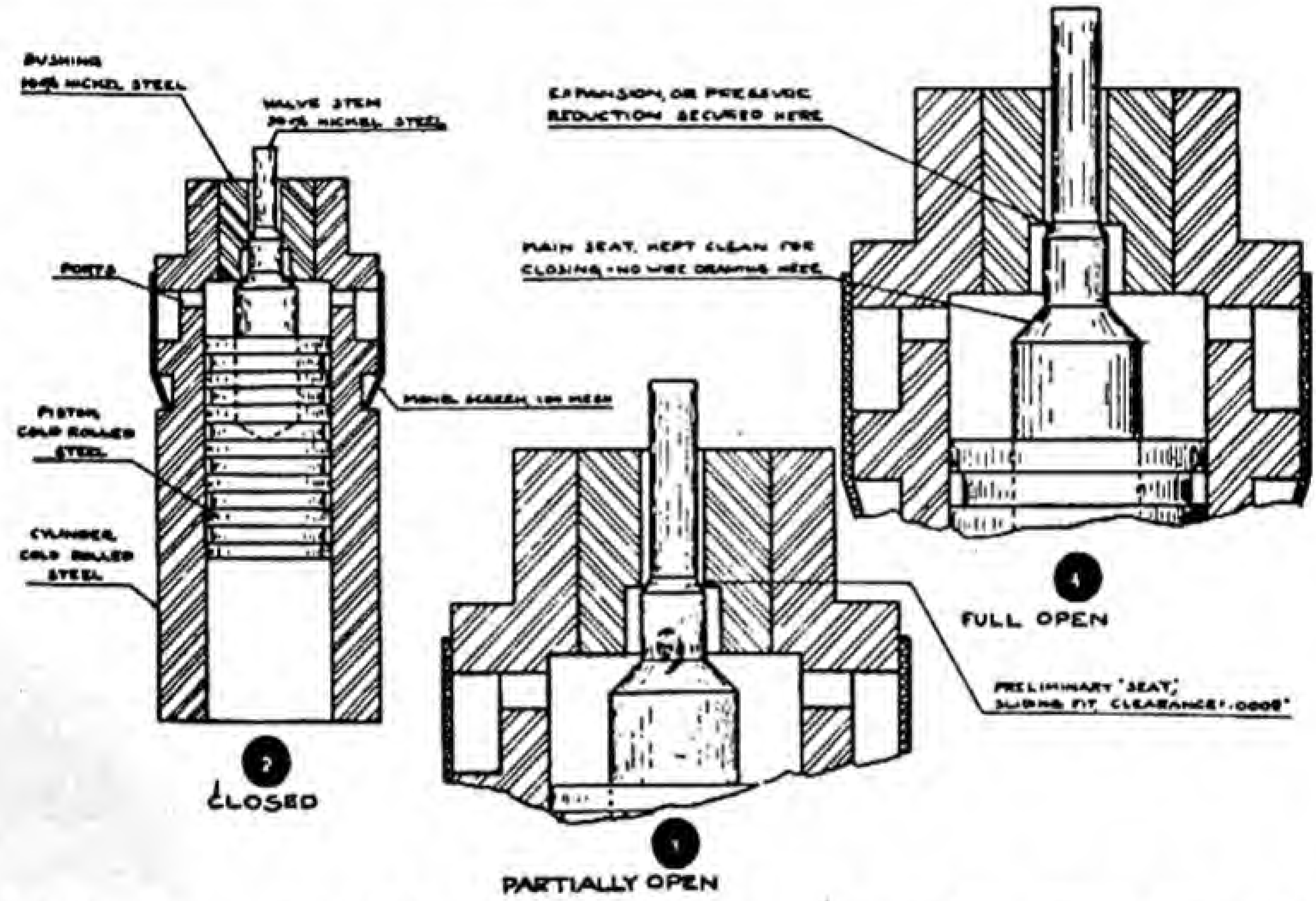


Fig. 124.—Expansion Valve (2) Closed; Expansion Valve (3) Partially Open; Expansion Valve (4) Fully Opened.

Table 68 shows the sizes and dimensions of this type of valve. The valve may be set for suction pressure varying from 0 to 30 lbs. The body, spring chamber, pressure plate, etc., are made of close-grained gray iron. The piston and cylinder are made of steel, but the piston seat and seat disc are made of 30 per cent nickel steel. The diaphragm and screen are made of Monel metal.

Liquid ammonia enters the valve on the inlet side and passes through the internal screen which collects all but the finest particles of foreign substance in the sediment chamber below. It then flows through a series of inlet ports and expands as the preliminary valve seat opens. Expanded, it enters the diaphragm chamber and through the outlet side into the delivery or expanded pressure line.

Assume that the initial pressure is 175 lbs. The required evaporation pressure is 25 lbs. As soon as the evaporation pressure exceeds

25 lbs. its action on the under side of the diaphragm will overcome the tension on the diaphragm spring, force the diaphragm up and permit the valve to close or to restrict itself to the point where proper expansion is effected. When the expansion pressure falls, the action of the diaphragm spring will force the valve open to the proper point to restore the required expansion.

There are, of course, no abrupt movements. In normal service, the valve has a floating motion, responding instantly in the very slightest changes in pressures.

To increase suction pressure, remove locking cap and screw down on adjusting screw. To decrease suction pressure release the diaphragm spring tension. For cleaning simply unscrew the bottom cap. All working parts are easily removed and after cleaning readily replaced.

The real work of throttling is performed at the smaller seat. The larger seat is doing practically no work.

At "4," Fig. 124, it is full open. The smaller seat is doing all the work; the larger seat none. Thus the larger seat is kept clean and unworn for tight closing.

No work will be done by the larger seat until the smaller seat is very badly worn from continued service. Only after the smaller seat is worn out and the larger seat does the work of throttling and becomes worn, will the unit need replacing. This preliminary seat unit in no way impairs the usual valve capacity.

Thermal Expansion Valves.—A sectional view of a thermostatic expansion valve is shown in Fig. 125. These valves are one of the most widely used automatic controls in refrigeration work. They regulate the flow of refrigerant to the evaporator in accordance with changes in superheat of the suction gas as it leaves the evaporator. They operate to maintain constant superheat.

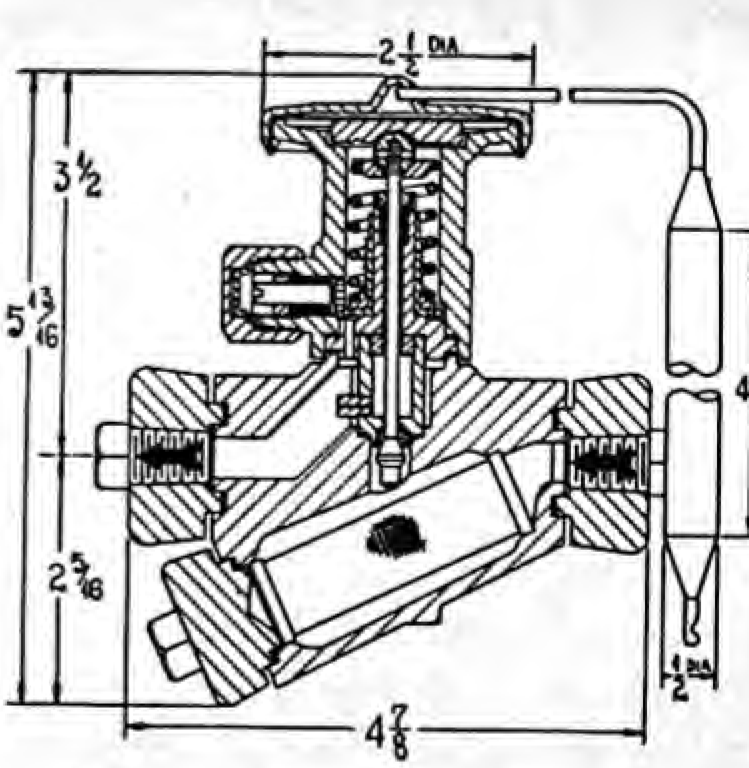


Fig. 125.—Alco Thermal Expansion Valve.

While the valve is called a thermal or thermostatic expansion valve it actually operates on changes in pressure. The bulb of the valve which is connected through a small diameter tube to the space above the diaphragm is charged with a volatile liquid. This bulb is either strapped to or inserted in the suction

line at the evaporator outlet. A rise in temperature of the suction gas evaporates the liquid in the bulb and the gas so formed exerts pressure above the diaphragm which tends to open the expansion valve. As the temperature of the suction gas drops, the gas in the bulb is cooled and finally condenses to lower the valve operating pressure and the spring closes the expansion valve. It is adjustable to maintain the desired number of degrees of superheat.

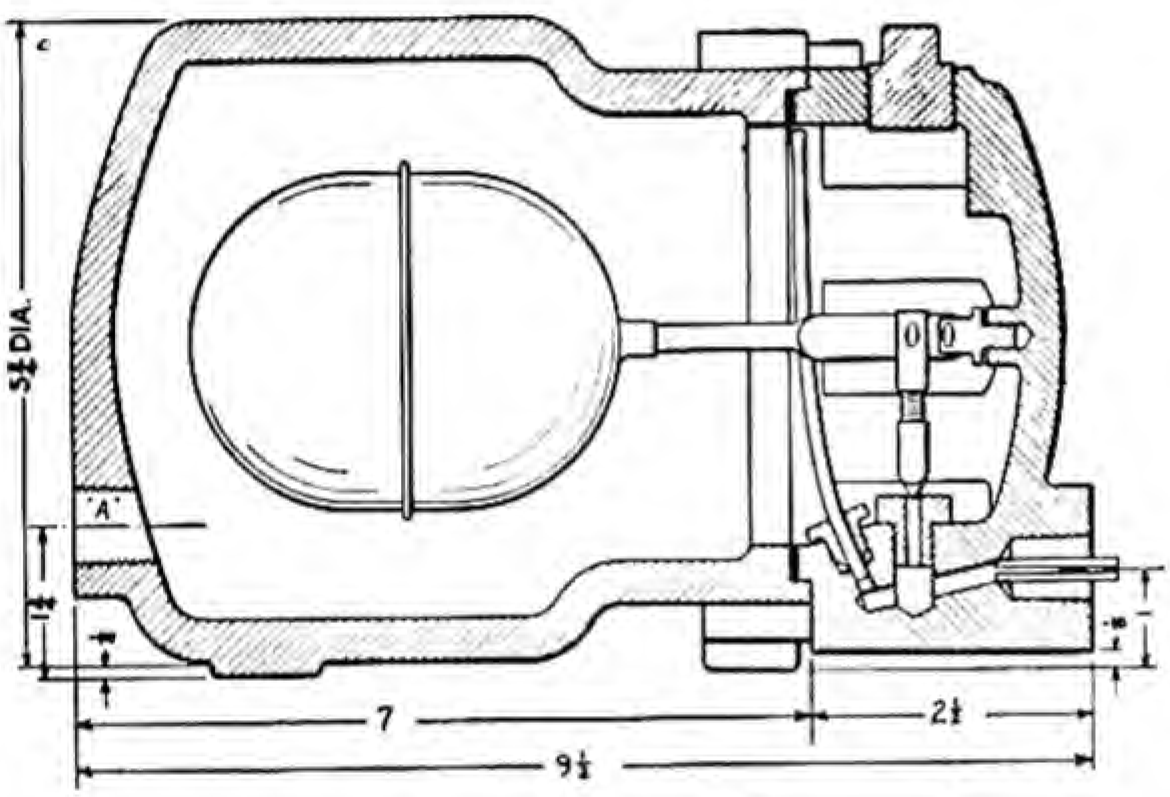


Fig. 126.—Alco Float Valve.

The thermal expansion valve finds its widest use in domestic and commercial refrigeration applications where the so called dry expansion type coil is used. It can also be applied, however, in connection with flooded systems.

Float Control Valve.—Float valves are sometimes used to control flow of liquid refrigerant in ice making and refrigeration. They are used quite extensively in the household machine. Float valves may be classified as high-pressure float valves or low-pressure float valves. High-pressure float valves are used to admit the refrigerant, as fast as it becomes liquefied, to the evaporating side of the system. Fig. 126 illustrates the construction of an Alco high-pressure float valve.

Low-pressure float valves are used to control flow of refrigerant into the evaporating unit. It is essential to properly design and lay out the flooded coil and connections so that a fairly constant liquid level may be established and which is really used to operate the float valve. It is also essential to lay out the coils so that the spent gas will

have a short travel, and so that the gas will pass to the suction header and leave the coil as soon as possible.

A flooded ammonia coil is illustrated in Fig. 127. This coil has a

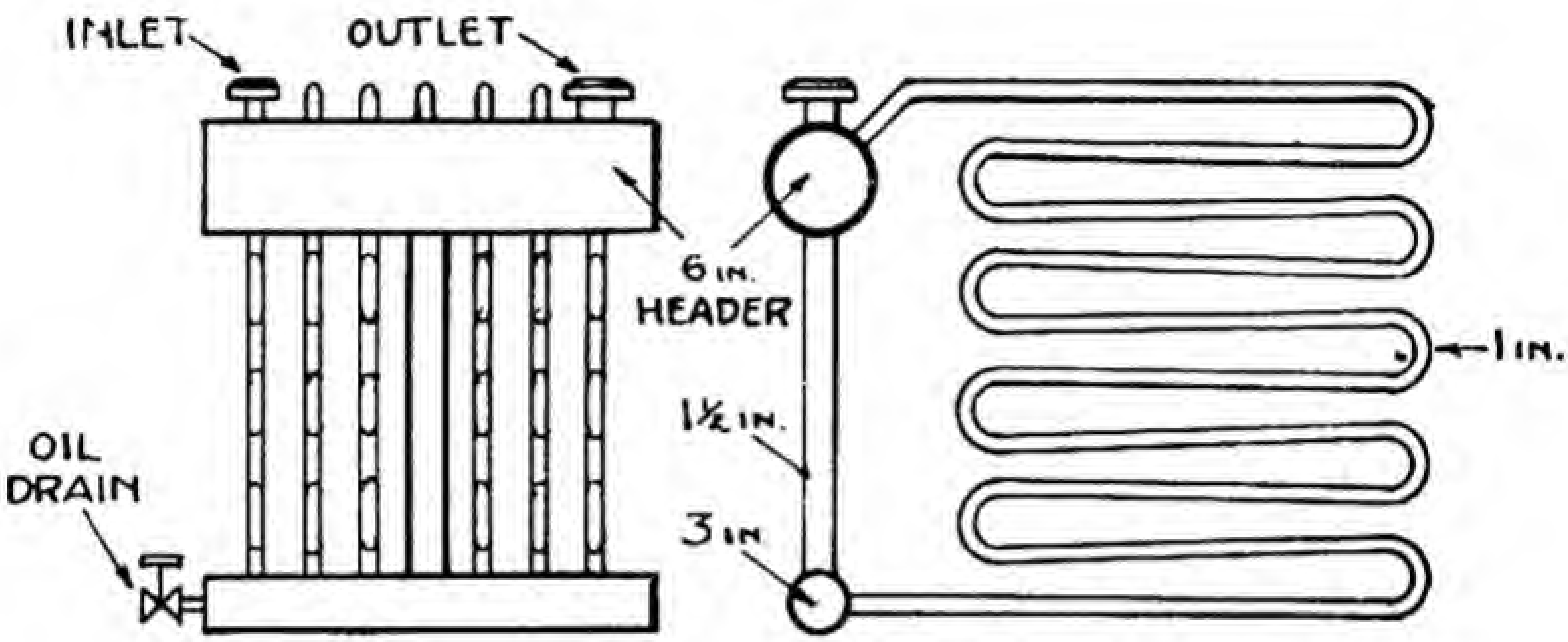


Fig. 127.—Flooded Ammonia Coil.

6-in. header at the top, which contains the connection for the inlet of liquid refrigerant and for the outlet of spent gases. The coil also has a lower liquid distributing header, which receives liquid from the top header and discharges same to the coil sections. Liquid refrigerant thrown from the coil is intercepted in the top header and returned to the coils through the lower header. The advantage of using the flooded principle lies in the fact that, if sufficient liquid is admitted, the coil surfaces are supplied with an excess of liquid refrigerant. The coil will then have a good heat transfer rate.

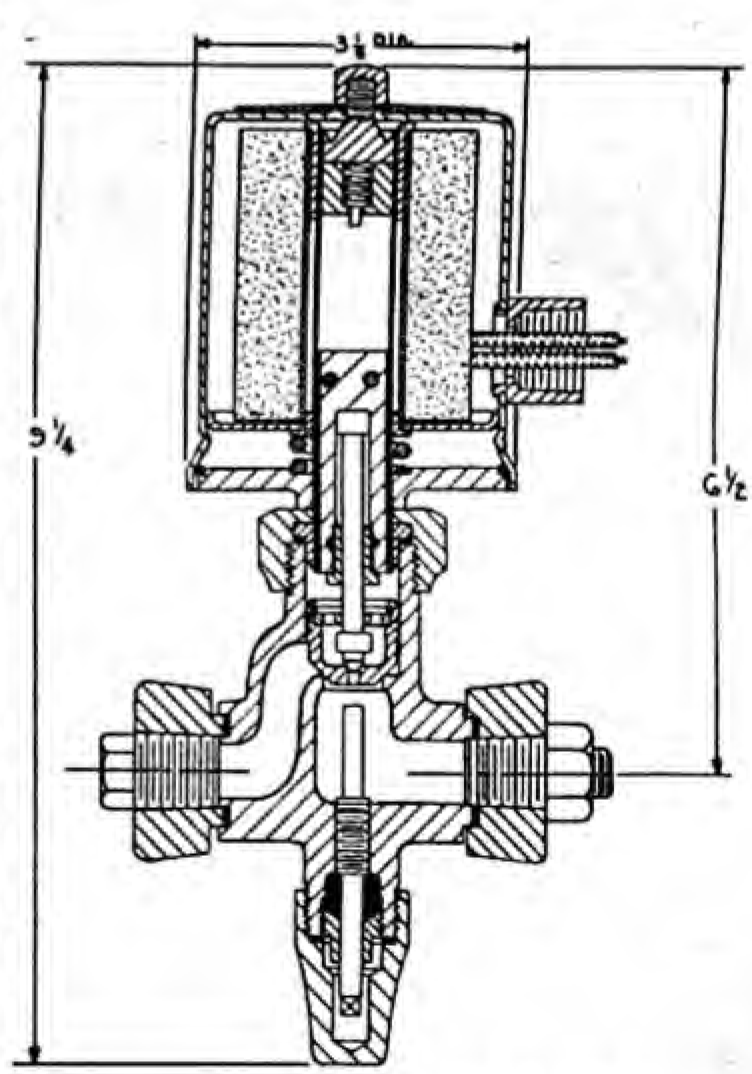


Fig. 128.—Magnetic Liquid Stop Valve.

Automatic Stop Valve.—When the compressor stops it is necessary on many automatic systems to stop the flow of refrigerant to the evaporator and also to stop the flow of gas in the suction line. When liquid feed is controlled solely by a thermal expansion valve the liquid may continue to flow

into the evaporator after the machine has stopped and result in slugs of liquid passing out in the suction gas when the compressor starts again.

A magnetic liquid stop valve is shown in Fig. 128. This valve is held open while the compressor is running and closes to stop liquid flow when the compressor stops. It is operated electrically through the solenoid at the top of the valve. The electrical connections to this solenoid are cut into the power circuit to the compressor motor.

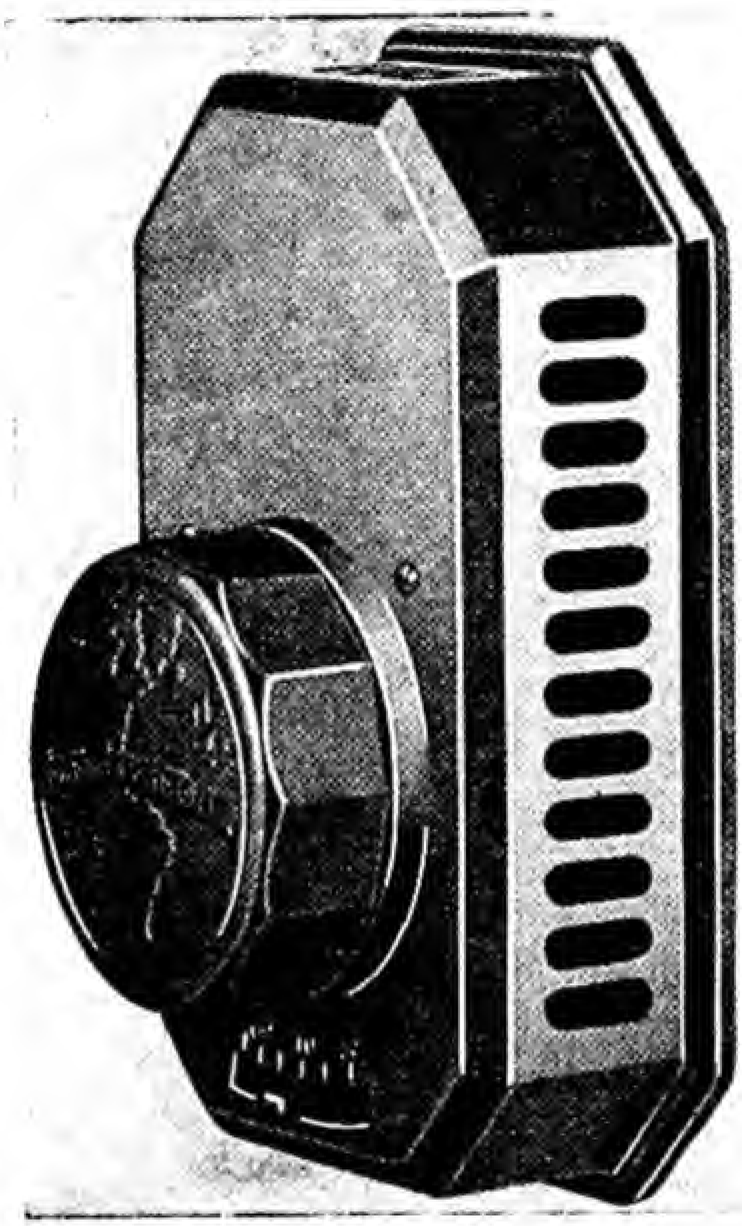


Fig. 129.—Mercoid Room Type Thermostat.

Suction stop valves are similarly applied to the suction end of the evaporator to cut it off from the system when the compressor stops. The suction stop valves may be solenoid operated like the liquid stop valves or they may be pressure operated from the liquid pressure just ahead of the expansion valve.

Thermostats.—Thermostats are commonly used on automatic plants to control temperatures by starting and stopping the main

refrigerating machine. These thermostats have their sensitive bulbs located in the space where it is desired to control the temperature. The electric circuit is made through a Mercoid bulb or an ordinary contact to the automatic motor controller, thereby starting and stopping the refrigerating machine in accordance with the temperature of the goods.

Thermostats may be classified as room thermostats, or remote control, or tank type thermostats. Fig. 129 illustrates a Mercoid room type thermostat. The operation is quite simple, as the power element or bellows expands or contracts through changes of temperature. The electric switch or mercury tube is then automatically thrown in the "on" or "off" position by means of a snap action mechanism.

Fig. 130.—Mercoid High Pressure Safety Cut-Out.

The remote control thermostat is equipped with a separate bulb and connecting tube. It is used for low temperature refrigeration work as well as for the control of the temperature of liquids, the sensitive bulb being located in the material to be controlled. In low temperature refrigeration work in rooms, the bulb is mounted in the room, and the main part of the thermostat is located outside of the room at a higher temperature. For control of liquids, the bulb is located in pipe lines, tanks, etc., as may be required.

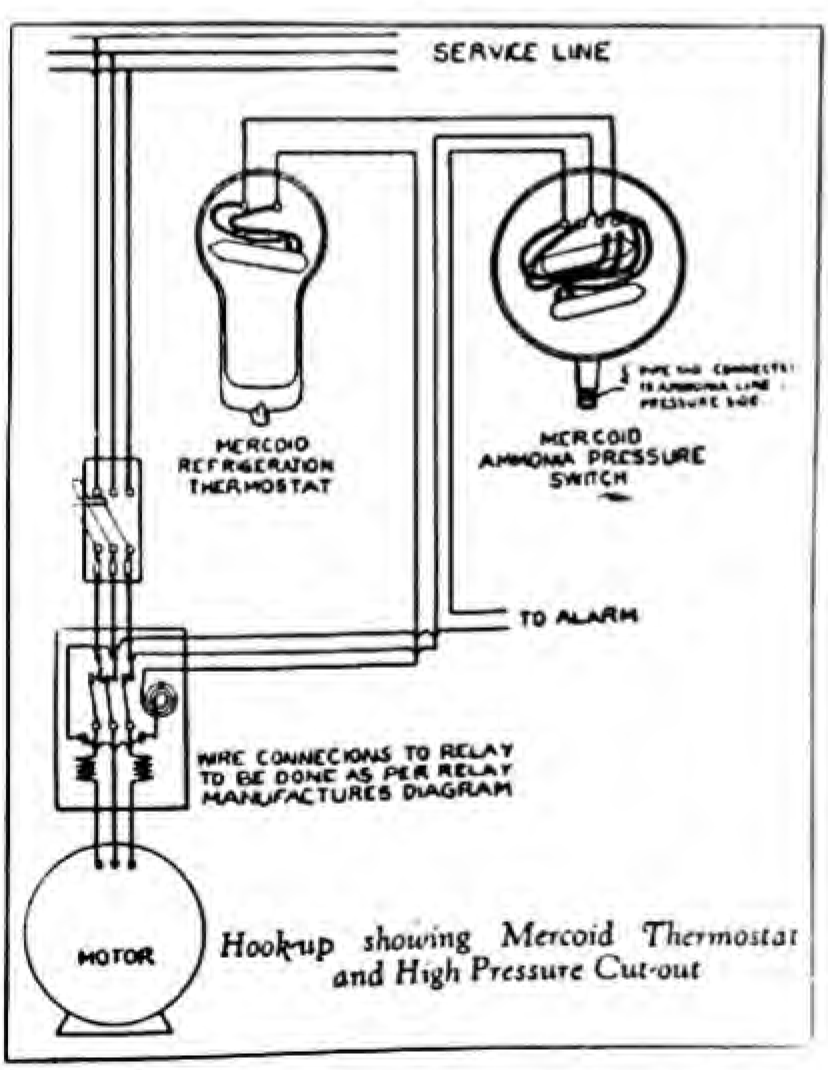


Fig. 131.—Schematic Wiring Diagram for Mercoid Thermostat and High-Pressure Cut-Out.

High-Pressure Cut-Outs.—High-pressure safety cut-outs are used to stop the motor running the refrigerating machine in the case of excessive pressure due to any cause. The high-pressure cut-off is connected to the automatic motor-starter so that it will stop the machine when the high-pressure exceeds a certain predetermined point. One type of high-pressure cut-out is shown in Fig. 130. A schematic wiring diagram for connecting a high-pressure cut-out, thermostat, and motor-starter is shown in Fig. 131.

QUESTIONS ON CHAPTER X.

1. What is the relationship of temperatures and pressures in evaporation?

2. What factors affect the amount of heat removed by refrigerants from evaporators?

3. Describe in detail the factors, enumerated in answer to Question 2, showing the effect of each factor, upon amount of heat removed from evaporator.

4. Ammonia evaporates at a pressure of 20 lbs. gauge; it condenses at a pressure of 175 lbs. gauge; it is subcooled to 65° F. before going to expansion valve; it leaves the evaporator at a temperature of 10° F. Find the heat absorbed in the evaporator by 1,000 lbs. of ammonia.

5. In Problem 4, what percentage of ammonia is evaporated at the expansion valve?

6. Describe the essential feature of the diaphragm type automatic expansion valve.

7. How much ammonia, expressed in lbs. and cu. in., must pass through automatic expansion valves, per min., on a 10-ton refrigeration plant operating under standard conditions?

8. Why is a liquid stop valve used when a thermostatic expansion valve is also in the system to control liquid feed?

9. Give three conditions in the system which might produce excess pressure and cause the high-pressure cut-out to stop the compressor motor.

10. Describe the essential parts of a completely automatic refrigerating system.

CHAPTER XI.

THE MANUFACTURE OF ICE.

Ice Making Systems.—Ice may be manufactured generally by means of two systems. In the first system the water to be frozen is placed in suitable sheet-iron cans, which are then nearly submerged in cold brine. The brine is maintained at a low temperature by the evaporation of a refrigerant in a suitable container. This is known as the can system of ice making. In the second system a coil or plate is immersed in the water to be frozen. The coils or plates are arranged for the evaporation of a suitable refrigerant, or for the circulation of cold brine. The removal of the heat from the water in this manner causes the formation of a layer of ice on each face of the coil or plate. This process is known as the plate system of ice making.

Can System.—The method of production of ice by means of the can system may be ascertained by an inspection of Fig. 132, which shows diagrammatically the principle of operation of this system. The water to be frozen is placed in suitable sheet-iron ice cans. The ice cans are then partly submerged in a cold brine, which is retained by a brine tank generally made of steel. The brine is maintained at a temperature of 10° to 18° F. by means of the evaporation of a suitable refrigerant such as ammonia or carbon dioxide. The water in the cans, being exposed to the pressure of the atmosphere, will freeze at a temperature of 32° F. The cold brine at a temperature such as 15° F. will therefore absorb heat from the water which is at a higher temperature. The brine will absorb the heat required to cool the water from the temperature of the supply to the freezing temperature, 32° F.; the latent heat of fusion of the water; the heat to cool the ice to the temperature of the brine; the heat from the ice cans; the heat transmitted by the insulation; the heat to cover the other losses. The transmission of heat from the air to the cold brine, of course, should be retarded as much as possible by using a suitable insulation on the brine tank. The brine is kept in constant circulation in the tank by means of a propeller agitator. The purpose of this is not only the equalization of the tem-

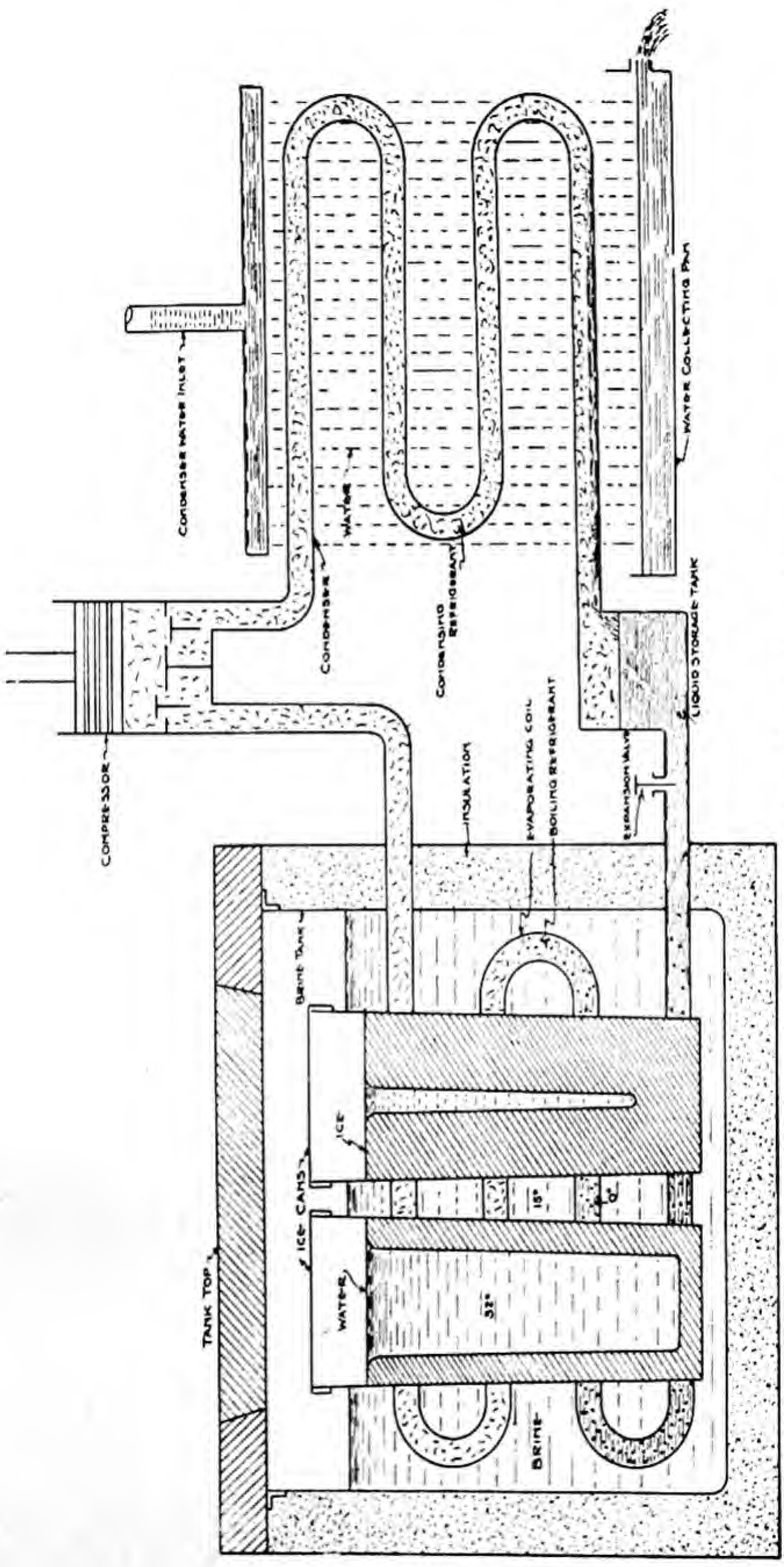


Fig. 132.—Diagram of Can Ice Making System.

perature of the brine in all parts of the tank, but also the increasing of the heat transmission which is due to the higher velocity.

As soon as the water in the cans is cooled to 32° F., ice will begin to form on the surfaces of the can. The rate of freezing is rapid at first, and becomes progressively slower as the thickness of the layer increases. If the can is left in the cold brine long enough, the water will entirely solidify, forming a homogeneous cake of ice. The cans of ice may then be removed from the brine and then other cans filled with water may be inserted in their places, thus making the process of ice production continuous.

Due to the fact that the brine absorbs heat as previously indicated, means must be provided for removing this heat just as fast as it is absorbed in order to maintain its temperature constant. This is usually accomplished by submerging an evaporator for a volatile refrigerant directly in the brine. This evaporator may be in the form of flat direct-expansion coils disposed between the rows of ice cans, as indicated by Fig. 132, or, it may consist of a suitable brine cooler of the shell-and-tube or double-pipe type.

The pressure upon the boiling refrigerant is maintained at such a point that the temperature is a few degrees below the temperature of the brine. For example, the refrigerant may have a temperature of 0° F., under which condition the heat in the brine at 15° F. will flow into the boiling refrigerant. The brine thus acts simply as a carrier of heat from the ice cans and ice tank to the evaporating refrigerant. The rest of the refrigerating is of the usual type involving the elements previously described.

It will be noted, therefore, that the formation of the ice is practically continuous and that the refrigeration system produces a uniform cooling effect.

Refrigeration Requirements.—The refrigeration required to produce ice under a given set of conditions depends, as previously indicated, upon the heat necessary to cool the water to the freezing point, to freeze it, to cool the ice to the temperature of the brine, and to cover other losses. Thus, if the initial temperature of the water is 70° F., the temperature of the brine is 12° F., and the unavoidable losses amount to 20 per cent of the actual refrigeration, the total refrigeration per pound of ice may be calculated as follows:

Heat to cool water (70° − 32°) x 1	= 38 Btu.
Latent heat of fusion	= 144 "
Heat to cool ice (32 − 12) x 0.5	= 10 "
	192 "
Unavoidable losses, 192 x 0.20	= 38.4 "
Total refrigeration	= 230.4 "

For this set of conditions one ton of ice would require approximately $\frac{230 \times 2000}{288{,}000} = 1.6$ tons of refrigeration. Assuming that the ice is cooled to 12° F. in each case, the following Table 69 will show how the tons of refrigeration per ton of ice will vary with the initial water temperature:

TABLE 69.—TONS OF REFRIGERATION PER TON OF ICE.

Initial Temp. of Water	Tons of Refr. per ton of Ice	Initial Temp. of Water	Tons of Refr. per ton of Ice
40	1.34	70	1.60
42	1.36	72	1.62
44	1.36	74	1.64
46	1.39	76	1.65
48	1.41	78	1.67
50	1.43	80	1.68
52	1.45	82	1.70
54	1.47	84	1.72
56	1.48	86	1.74
58	1.50	88	1.76
60	1.52	90	1.77
62	1.53	92	1.79
64	1.55	94	1.81
66	1.57	96	1.82
68	1.58		

Freezing Time of Ice.—The freezing time or rate for a given thickness of ice as compared to the freezing time of ice of another thickness varies approximately as the square of the thickness, other factors remaining constant. In other words, the freezing time required to freeze ice 4 inches and 6 inches thick varies in the ratio 16 to 36, also, the freezing time is inversely proportional to the temperature between the brine and the freezing water.

For example, in a given ice tank freezing time required to freeze a 11-in. block of ice was found to be 47 hours with 16° F. brine. The question is—what would be the freezing time if the brine temperature is reduced to 12° F., all other factors remaining the same? This would be found as follows:

$$\frac{47 \times (32 - 16)}{32 - 12} = 37.6 \text{ hours.}$$

The freezing time of ice depends also upon such factors as the rate and efficiency of brine circulation, the layout and design of the ice tank, etc.

The older formula for the freezing time is as follows:

$$\text{f.t.} = \frac{7 \times t^2}{32 - t_b} \qquad (1)$$

Where t equals thickness in inches of ice at top of block, t_b equals temperature of brine. From data taken from modern ice tanks, with good circulation and layout of tank, operated under what might be termed standard practice, it was found that the constant 7 in the preceding formula should be reduced to 6.2 or lower. The formula, connecting freezing time, brine temperature, and thickness of ice block, can now be written as follows:

$$\text{f.t.} = \frac{6.2 \times t^2}{32 - t_b} \qquad (2)$$

Example.—What is the freezing time in an ice tank when the brine temperature is 14° F. and the thickness of the ice block is 11½ inches?

Solution:

$$\text{f.t.} = \frac{6.2 \times 11.5 \times 11.5}{32 - 14} = 45.6 \text{ hours.}$$

Table 70 gives the freezing time of can ice for various thicknesses of ice and various brine temperatures under average conditions.

In order to obtain the best freezing time, the ice cans should be submerged in the brine to the proper depth. The top of the finished ice block should never be above the brine level. It should be 1 in. or more below the brine level.

Table 71 gives the number of cans per ton of ice making capacity for various brine temperatures and sizes of cans. The newer plants will have a lower can ratio but many of the older plants operate in the range shown.

Cans Per Ton of Ice.—The number of ice cans required per ton of ice per day may be expressed as follows:

$$N = \frac{\frac{2000}{24\,W}}{\text{f.t.}} \qquad (3)$$

where N = cans per ton of ice
W = weight of ice blocks, lbs.

Transposing the above formula, another relation is obtained for the freezing time:

$$\text{f.t.} = \frac{24\,N\,W}{2000} \qquad (4)$$

The above formula is obtained from the simple arithmetic of the problem.

TABLE 70.—FREEZING TIME FOR CAN ICE—HOURS.

Brine Temp. Degrees Fahr.	THICKNESS OF ICE IN INCHES									
	4	5	6	7	8	9	10	11	11½	12
4	3.57	5.58	8.03	11.0	14.3	18.1	22.3	27.0	29.5	32.1
5	3.70	5.78	8.33	11.4	14.8	18.8	23.2	28.0	30.6	33.3
6	3.84	6.01	8.65	11.8	15.4	19.5	24.0	29.1	31.8	34.6
7	4.00	6.25	9.00	12.3	16.0	20.3	25.0	30.5	33.1	36.0
8	4.17	6.51	9.37	12.8	16.6	21.1	26.0	31.5	34.4	37.5
9	4.35	6.79	9.78	13.3	17.4	22.0	27.2	32.9	36.0	39.1
10	4.55	7.10	10.30	14.0	18.2	23.0	28.4	34.4	37.6	40.9
11	4.76	7.44	10.70	14.6	19.0	24.1	29.8	36.0	39.3	42.8
12	5.00	7.81	11.20	15.4	20.0	25.3	31.3	37.8	41.3	45.0
13	5.26	8.18	11.80	16.2	21.1	26.7	32.9	39.8	43.5	47.4
14	5.55	8.63	12.50	17.0	22.2	28.2	34.7	42.0	45.8	50.0
15	5.88	9.18	13.20	18.0	23.5	29.8	36.8	44.5	48.6	52.9
16	6.25	9.76	14.00	19.2	25.0	31.7	39.0	47.3	51.7	56.3
17	6.66	10.40	14.90	20.4	26.6	33.8	41.7	50.4	55.1	60.0
18	7.14	11.10	16.00	21.9	28.5	36.2	44.6	54.1	59.1	64.3
19	7.68	12.00	17.30	23.6	31.5	38.9	48.1	58.2	63.6	69.3
20	8.33	13.00	18.70	25.6	33.3	42.2	52.1	63.0	68.7	75.0
21	9.09	14.20	20.5	27.9	36.2	46.1	56.8	68.8	75.2	81.8
22	10.00	15.6	22.5	30.6	40.0	50.7	62.5	75.7	82.7	90.0
23	11.10	17.3	25.0	34.0	44.5	56.4	69.5	84.0	91.8	100.0
24	12.50	19.5	28.1	38.3	50.0	63.5	81.0	94.6	103.5	112.5

TABLE 71.—NUMBER OF CANS PER TON OF ICE AT VARIOUS BRINE TEMPERATURES.

Brine Temp. Degrees Fahr.	SIZE OF ICE CANS							
	50 Lbs. 5" x 14"	50 Lbs. 6" x 12"	100 Lbs. 8" x 16"	200 Lbs. 11" x 22"	300 Lbs. 11" x 22"	300 Lbs. 11½" x 22½"	400 Lbs. 11" x 22"	400 Lbs. 11½" x 22½"
4	9.30	13.3	11.9	11.2	7.50	8.19	5.62	6.15
5	9.63	13.8	12.3	11.6	7.78	8.50	5.83	6.37
6	10.10	14.4	12.8	12.1	8.08	8.83	6.06	6.62
7	10.40	15.0	13.3	12.7	8.47	9.19	6.36	6.89
8	10.80	15.6	13.8	13.1	8.75	9.55	6.56	7.16
9	11.30	16.7	14.5	13.7	9.13	10.0	6.85	7.50
10	11.70	17.1	15.1	14.3	9.56	10.4	7.16	7.84
11	12.40	17.8	15.8	15.0	10.0	10.9	7.50	8.19
12	13.00	18.6	16.7	15.7	10.5	11.5	7.88	8.61
13	13.60	19.6	17.6	16.5	11.1	12.1	8.29	9.06
14	14.20	20.8	18.5	17.5	11.6	12.7	8.75	9.54
15	15.30	22.0	19.6	18.6	12.3	13.5	9.27	10.1
16	16.30	23.3	20.8	19.7	13.1	14.3	9.85	10.7
17	17.40	24.8	22.2	21.0	14.0	15.3	10.5	11.5
18	18.50	26.7	23.7	22.5	15.0	16.4	11.2	12.3
19	20.00	28.8	26.2	24.2	16.2	17.6	12.1	13.2
20	21.60	31.1	27.8	26.2	17.5	19.1	13.2	14.3
21	23.70	34.1	30.1	28.6	19.1	20.9	14.3	15.6
22	26.00	37.5	33.3	31.5	21.0	23.0	15.7	17.2

By comparing the formulas 2 and 4, the following formula for the required number of ice cans is obtained:

$$N = \frac{516.6 \times t^2}{W\ (32 - t_b)}$$

Example.—How many ice cans are required for the following conditions:

Brine temp. equals 14° F.
Weight of ice block equals 400 lbs.
Thickness of ice block equals 11 in.

Solution:

$$N = \frac{516.6 \times 11.5 \times 11.5}{400 \times (32 - 14)}$$

equals 9.5 cans per ton of ice per day.

The above formula may be worked backwards to check up the operation of an ice tank in an actual plant.

Example.—Weight of ice blocks equals 300 lbs.

Brine temp. equals 15° F.
Cans per ton of ice produced equals 11.4
Thickness of cans equals 11 in.

Solution:

$$11.4 = \frac{X \times 11 \times 11}{300 \times (32 - 12)}$$

X equals 555 or correct number of cans

$$N = \frac{516.6 \times 11 \times 11}{300 \times (32 - 12)} \text{ equals } 10.4$$

The relative submergence of the ice cans has an important bearing upon the freezing time of the ice as well as the shape of the top of the ice block. Generally speaking, the higher the finished ice level is above the brine, the longer the freezing time. Under ordinary conditions, it has been found that the best submergence is to have the top of the finished ice block about 1½-in. below the level of the brine.

Table 72 gives the various dimensions, weights, and specifications of various sizes of ice cans.

As previously indicated, the relation of the number of cans and the freezing time is expressed by the following formula:

$$f.t. = \frac{N \times W \times 24}{2000}$$

where f.t. = freezing time
N = no. of cans per ton of ice
W = weight of blocks of ice in lbs.

TABLE 72.—SPECIFICATIONS OF ICE CANS.
York Ice Machinery Corp.

Capacity of Can in Pounds	Inside Dimensions Top	Inside Dimensions Bottom	Length	Length Over All	Thickness of Galv. Steel U. S. Std. Gauge	Freezing System	Type of Cake	Weight of Can Pounds
50	6" x 12"	5½" x 11½"	28"	29"	16	Individual	Solid	34
50	8" x 8"	7¼" x 7¼"	31"	32"	16	Individual	Solid	27
50	5" x 14"	4¼" x 13¼"	31"	32"	16	Individual	Solid	33
50	4" x 16"	3¼" x 15¼"	31"	32"	16	Individual	Solid	35
100	8" x 16"	7¼" x 15¼"	31"	32"	16	Individual	Solid	40
200	11½" x 22½"	10½" x 21½"	31"	32"	16	Individual	Solid	60
304	11½" x 22½"	10½" x 21½"	44"	45"	16	Individual	Solid	76
324	11½" x 22½"	10½" x 21½"	46"	47"	16	Individual	Scored	80
304	11½" x 22½"	10½" x 21½"	46"	47"	16	Group	Solid	80
324	11½" x 22½"	10½" x 21½"	48"	49"	16	Group	Scored	83
410	11½" x 22½"	10½" x 21½"	57"	58"	14	Individual	Solid	108
430	11½" x 22½"	10½" x 21½"	59"	60"	14	Individual	Scored	112
410	11½" x 22½"	10½" x 21½"	59"	60"	14	Group	Solid	112
430	11½" x 22½"	10½" x 21½"	62"	63"	14	Group	Scored	116
304	11" x 22"	10" x 21"	47"	48"	16	Individual	Solid	80
324	11" x 22"	10" x 21"	49"	50"	16	Individual	Scored	83
304	11" x 22"	10" x 21"	49"	50"	16	Group	Solid	83
324	11" x 22"	10" x 21"	51"	52"	16	Group	Scored	86
410	11" x 22"	10" x 21"	61"	62"	14	Individual	Solid	111
430	11" x 22"	10" x 21"	63"	64"	14	Individual	Scored	115
410	11" x 22"	10" x 21"	63"	64"	14	Group	Solid	115
430	11" x 22"	10" x 21"	66"	67"	14	Group	Scored	120

Substituting 50, 100, 200, 300 and 400 for the value of W, the formula becomes as follows:

for 50 lb. cans, f.t. = 0.6 N
" 100 " " f.t. = 1.2 N
" 200 " " f.t. = 2.4 N
" 300 " " f.t. = 3.6 N
" 400 " " f.t. = 4.8 N

The determination of the correct number of cans per ton of ice is a question of economic importance. If too few cans are installed, a very low brine temperature is required in order to freeze the ice in the required time. This, of course, means a very low suction pressure on the machine. Again, if too many cans are put into the tank per ton of ice, the initial cost becomes too great. But the better working conditions are approached with the larger amount of cans, since a higher suction pressure may be carried generally. To decide this economic question it therefore is necessary to consider the initial cost of ground, building and equipment, together with the cost of power required to operate the plant under the given conditions.

Under general conditions the correct brine temperature may be said to vary from 10° to 20° F. This would mean that the number of 300-lb. 11-in. standard ice cans per ton of ice would vary from 9.56 to 17.5.

As a general rule thirteen 300-lb. cans may be used per ton of ice made. The exact number of ice cans to be used in a given plant will depend upon local conditions.

Brine Coolers.—Various forms of apparatus may be adopted for absorbing the heat from the brine. These apparatus are generally suitable containers for holding the evaporating refrigerant. Such forms as flat direct-expansion pipe coils, shell-and-tube brine coolers, and double-pipe brine coolers, etc., may be used. The direct-expansion coils and the double-pipe brine coolers may be operated on either the dry direct-expansion system or the flooded system. The shell-and-tube brine cooler in itself may be considered as operating on the flooded system.

The desirable arrangement of these evaporating surfaces has been considered in a previous Chapter on the methods of distribution of refrigeration. It will be interesting to note how the amount of surface of these apparatus will vary with different conditions. In general, the amount of surface to be installed depends upon the refrigeration to be produced per ton of ice, the heat transfer coefficient of the surface, and the temperature difference between the boiling refrigerant and the brine.

It is evident, in consideration of the amount of evaporating surface to be installed, that the exact amount of refrigeration per ton of ice must be estimated. The values in Table 69 have been estimated with losses equivalent to 20 per cent of the actual refrigeration. This value, of course, will cover the losses under ordinary conditions and it is evident that any other additional refrigeration must be added to this amount. Sometimes the ice storage room may be refrigerated by brine from the ice tank. The heat transfer coefficients for various evaporator surfaces may be taken from Table 60 of Chapter VIII.

The mean temperature difference between the boiling refrigerant and the brine is determined by economic considerations, and will vary from 6° to 12° F. If a small temperature difference is used it is necessary to install a large amount of surface. This, of course, will give a high suction pressure, but the initial cost of the evaporating surface becomes too great. On the other hand, if the temperature difference is quite large, a small amount of evaporating surface may be installed, and under this condition the suction pressure may drop to a fairly low level. Lowering the suction pressure, of course, increases the horsepower per ton of refrigeration. Thus it will be observed that the magnitude of the temperature difference will depend upon such factors as the cost of the equipment, the cost of producing a ton of refrigeration, and the local conditions. By referring to Table 69 it will be observed that if water at a temperature of 70° F. is available for making ice, the

TABLE 73.—DIRECT EXPANSION PIPE (1¼ IN.) PER TON OF ICE.

Brine Temp. Deg. F.	Refrigerant Temperature, deg. Fahr. −4°	−2°	0°	2°	4°	6°	8°	10°	12°
4	30	40	60	120					
6	24	30	40	60	120				
8	20	24	30	40	60	120			
10	17.2	20	24	30	40	60	120		
12	15	17.2	20	24	30	40	60	120	
14	13.3	15	17.2	20	24	30	40	60	120
16	12	13.3	15	17.2	20	24	30	40	60
18	10.9	12.0	13.3	15	17.2	20	24	30	40
20	10	10.9	12.0	13.3	15	17.2	20	24	30
22		10	10.9	12.0	13.3	15.0	17.2	20	24
24			10	10.9	12.0	13.3	15.0	17.2	20

TABLE 74.—SHELL-AND-TUBE AND DOUBLE-PIPE BRINE COOLER SURFACE IN SQ. FT. PER TON OF ICE.

Brine Temp. Deg. F.	Refrigerant Temperature, deg. Fahr. −4°	−2°	0°	2°	4°	6°	8°	10°	12°
4	368	492	737	1474					
6	295	368	492	737	1474				
8	245	295	368	492	737	1474			
10	210	245	295	368	492	737	1474		
12	184	210	245	295	368	492	737	1474	
14	164	184	210	245	295	368	492	737	1474
16	148	164	184	210	245	295	368	492	737
18	134	148	164	184	210	245	295	368	492
20	123	134	148	164	184	210	245	295	368
22	114	123	134	148	164	184	210	245	295
24	105	114	123	134	148	164	184	210	245

required refrigeration for a ton of ice will be equal to 1.6 tons. The number of lineal feet of 1¼-in. direct-expansion pipe to be installed per ton of ice may be calculated as follows:

$$\text{Feet of 1¼-in. pipe per ton} = \frac{288{,}000 \times 1.6 \times 2.301}{15 \times 24 \times t_d}$$

where t_d = temperature difference

The amounts of direct expansion pipe per ton of ice, according to the above formula, have been calculated and tabulated in Table 73. The amounts of direct expansion coil surface which will give temperature differences varying from 6° to 12° F. are included between the heavy lines.

In a similar manner, the number of square feet of double-pipe brine cooler and shell-and-tube brine cooler surface per ton of ice may be calculated by the following formula:

$$\text{Sq. ft. of surface} = \frac{288{,}000 \times 1.6}{80 \times 24 \times t_d}$$

where t_d = temperature difference

The amounts of surface per ton of ice for different temperatures of brine and refrigerant have been calculated and tabulated in Table 74. The surfaces corresponding to temperature differences of 6° to 12° F. are included between the heavy lines.

From an inspection of Tables 73 and 74 it will be observed that the amount of surface increases rapidly as the temperature differences are reduced. The advantage of installing the larger amount of surface is likewise diminished, so that it is advisable to install such amounts of surface which are included between the heavy lines in the table.

Table 73 is based on a heat transfer coefficient of 15. Table 74 is based on a heat transfer coefficient of 80. If other coefficients are used, then the amount of surface will vary directly as the coefficients.

Ice Freezing Tanks.—Ice freezing tanks are generally constructed of steel, although wood and concrete are sometimes used for such tanks. Due to the fact that the temperatures in brine tanks are fairly low, it is usually beneficial to insulate the freezing tanks in an efficient manner. The bottoms of these tanks are generally insulated by applying five to six inches of insulation in the board form. The sides and ends of the tank may be insulated by applying five or six inches of board insulation, or by using ten or twelve inches of granulated insulating material. The steel ice freezing tanks, of course, have a larger initial cost, but will probably give the lowest cost over a period of years, at the same time eliminating difficulties due to leaks, etc.

The ice cans are supported in the tank by means of a suitable framework of wood. In some cases the cans rest upon timbers which lie upon the bottom of the tank. In other constructions the cans are supported by the framework on the top of the tank.

Suitable movable covers made of wood are provided for the ice cans. Covers are generally made of oak or cypress. The arrangement of the framework, can covers, etc., in a raw water ice making tank may be ascertained by inspecting Fig. 133. In this type of construction the tank top is supported by heavy stringers of timber which extend from one side of the tank to the other. These stringers, in turn, are supported at several points by struts which rest upon the bottom of the tank. The cans are inserted in the tank through the various compartments of the tank top. In Fig. 133 a single cover is provided for every two cans. However, it is ordinarily advisable to use can covers for 4 or 5 cans.

Ice making tanks generally have a length which is two to three times the width. The depth of the tank is determined principally by the depth of the ice cans, and in general is very near the depth of the ice cans, varying one or two inches as the case may be. The tanks may be made in such sizes as to produce as high as one hundred tons of ice per day. However, the average ice plant will operate more flexibly by using freezing tanks which contain ice cans for making 50 to 60 tons of ice per day. The larger ice tanks will have from 24 to 30 cans in width, while the number of cans in the length of the tank may be as many as 42 to 48. Of course, the exact layout of the ice making tank depends, in most cases, upon the local conditions in the plant.

In order to produce an even brine temperature in all parts of the tank and in order to increase the heat transfer coefficient of the evaporating surface it is generally advisable to produce a rapid circulation of the brine throughout all parts of the tank. This is generally accomplished by dividing the tank into compartments by means of partitions and bulkheads. A propeller agitator is used to cause the brine to circulate rapidly in the tank. These agitators may be arranged either in a vertical or horizontal position and should be of sufficient size to cause the brine to circulate throughout the whole tank in a few minutes. Ordinarily, the speed of the brine circulation should not be less than 25 ft. p. m.; 35 ft. p. m. is preferable. Especial attention should be given to securing an even distribution of brine flow in all parts of the tank.

The arrangement of a small ice making tank containing bulkheads, partitions, horizontal agitator, and a shell brine cooler is shown in Fig. 134. In this figure the brine circulates through the tank in two different circuits. The arrangement of the agitator, bulkheads, partitions, etc., is also shown.

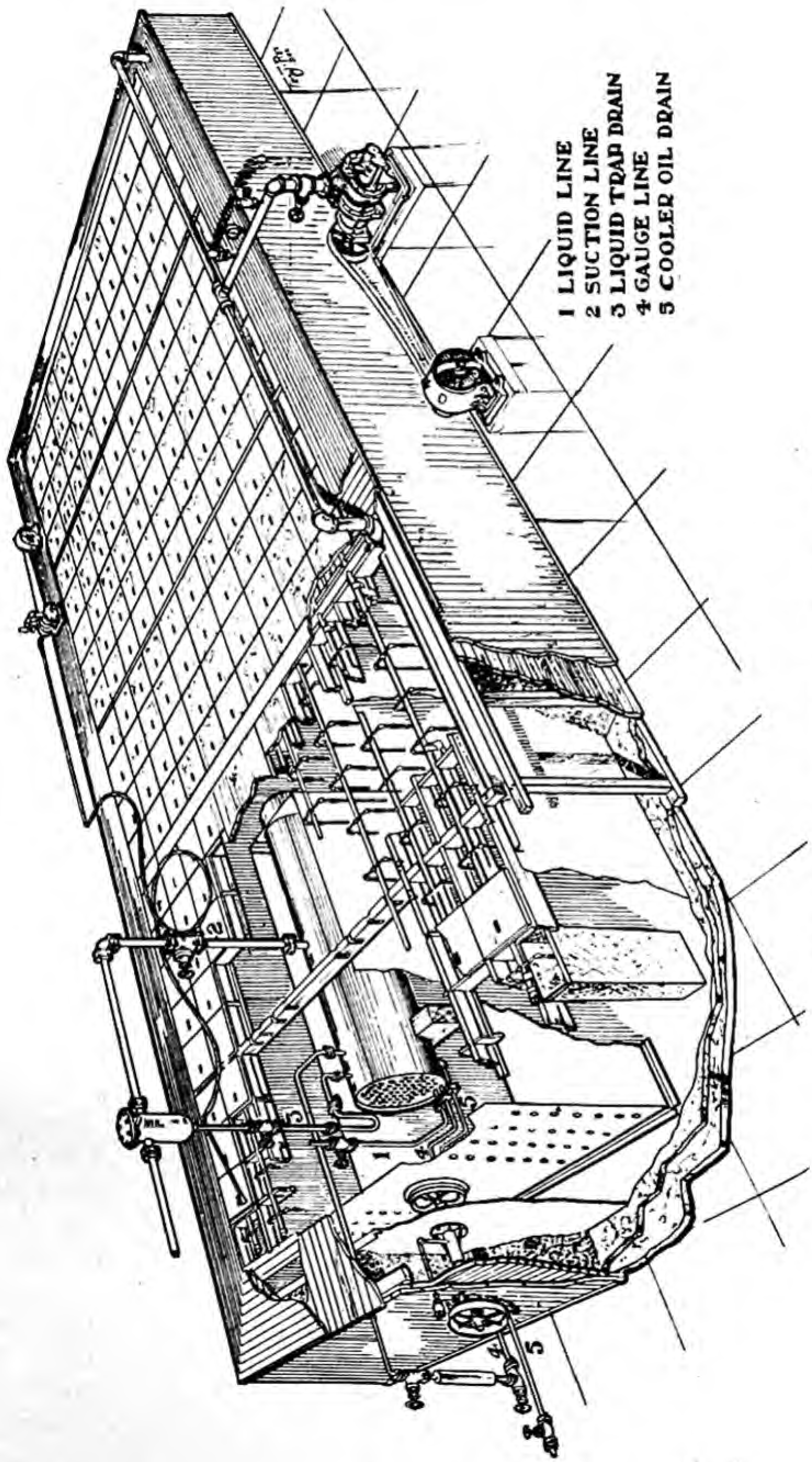


Fig. 133.—Arrangement of Ice Making Tank.

The refrigeration to produce the required quantity of ice and to offset the other heat losses is produced generally by the direct evaporation of a liquid refrigerant. Vertical sections of pipe coils, placed between the rows of ice cans, are generally used for this purpose. These pipe coils are connected to suitable gas and liquid headers at the end of the tank. They, of course, may be operated on either the dry expansion or the flooded system. A shell-and-tube brine cooler which is submerged directly in the brine in the tank is sometimes used. The arrangement of such a cooler is shown by Fig. 133. The brine, in circulating about the tank, is warmed a fraction of a degree, and by being forced through the shell brine cooler it is in turn cooled the same amount, so that the brine acts as a carrier of heat.

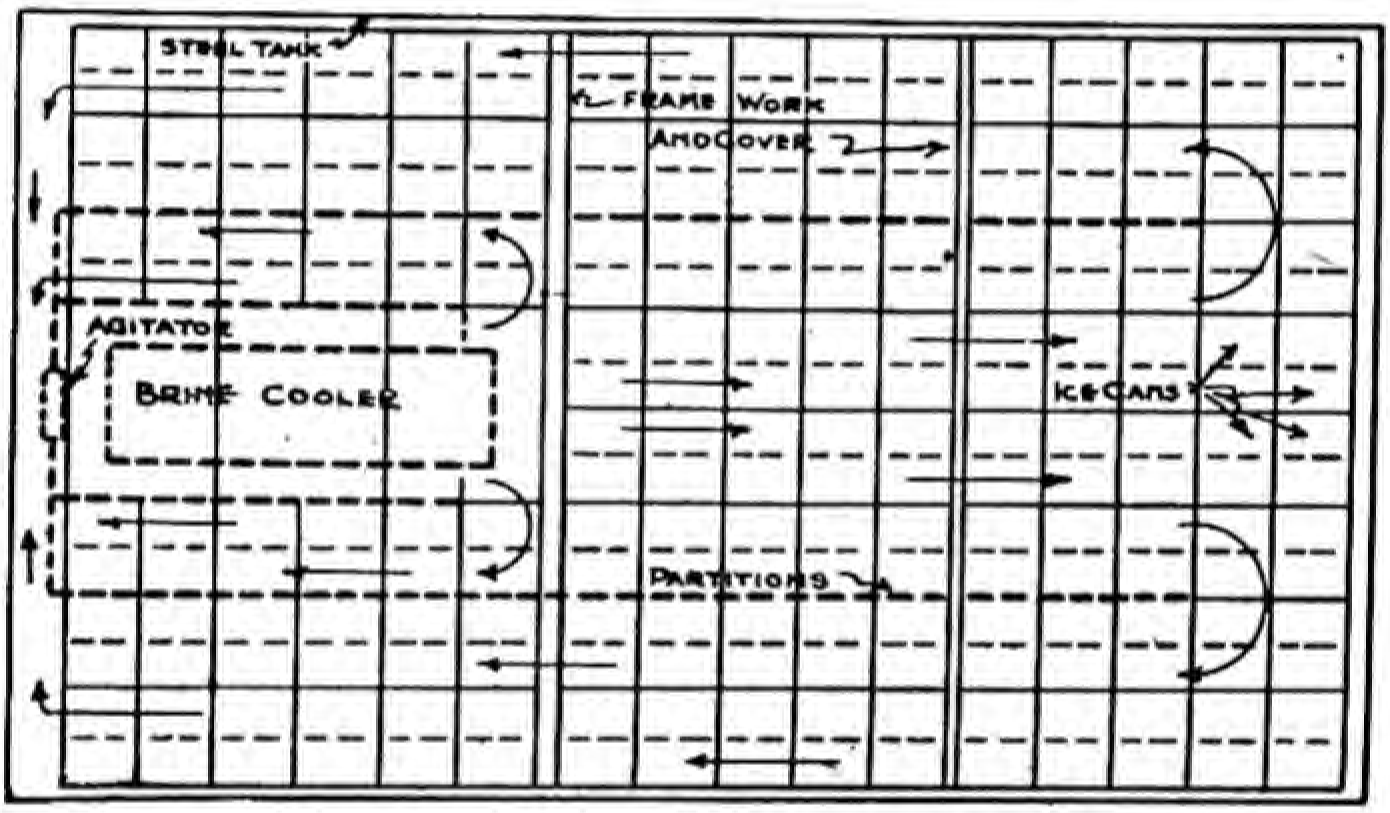


Fig. 134.—Freezing Tank Layout.

The water to be frozen is placed in the ice cans, which are in turn nearly submerged in the cold brine in the ice tank. After the water in the cans has entirely solidified, the cans are removed from the tank. This is generally accomplished by a suitable overhead traveling crane which lifts the cans and ice from the tank. Any number of ice cans, from one to twenty-four, may be lifted at once. After the cans have been lifted from the cold brine, they are taken by the traveling crane to the end of the tank to a thawing device. In some cases the cold ice cans are submerged in water for the purpose of loosening the ice in the cans. In other plants the cans are sprayed with water to accomplish the same purpose. After the ice has been slightly thawed in the cans, the cans are placed in a suitable dumping apparatus, which, by tilting the can over to a horizontal position, allows the ice to slide out of the can through a door into a storage room.

After the ice has been removed from the cans, the cans are again filled with the water to be frozen. In some cases this is accomplished by inserting an automatic can filler into the cans, which allows the water to flow into the cans until a certain predetermined height has been reached. This method of filling the cans is used after the cans have been replaced in the cold brine. Another method of filling the cans consists of a measuring tank which is located near the ice dumping apparatus. When the ice cans are returned to a vertical position in the ice dumping apparatus, just after the ice has been removed, the ice cans may be refilled with water by the manipulation of valves and connections to the measuring tank. By this means a uniform amount of water may be put into the cans at each refilling.

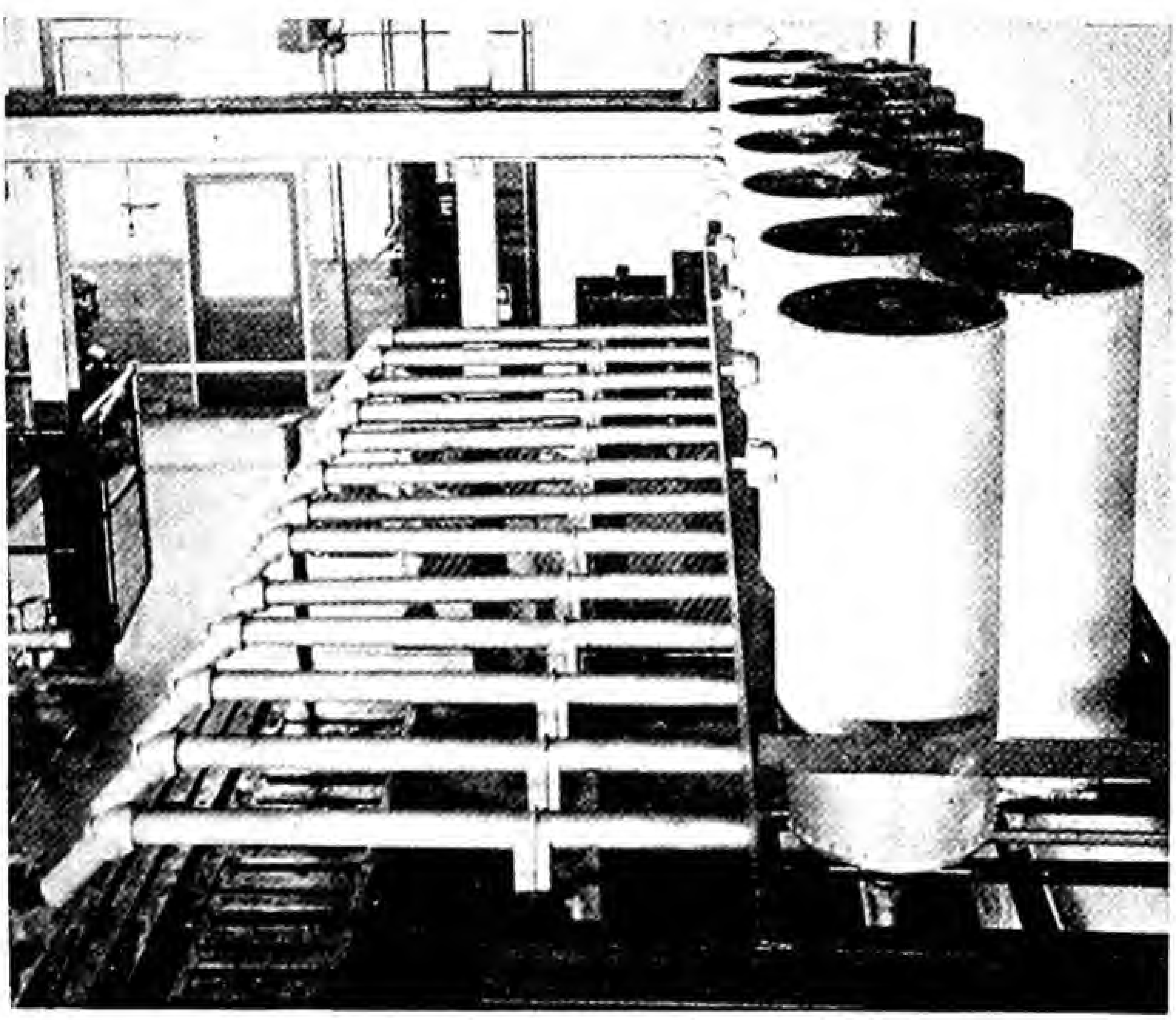


Fig. 135.—Multiple Can Filling Tanks.

Brine Circulators.—Brine circulators, or agitators, are used to cause the brine to flow around the ice cans and over the brine cooling surfaces, such as coils, or shell-and-tube brine coolers. Brine circulators may be either the vertical or horizontal types. Ordinarily, the horizontal circulators are belted to the motors which are used for driving them. The vertical circulators are usually direct-connected to suitable electric motors. The use of the vertical circulator eliminates the use

of a stuffing box in the side wall of the tank, which is of course necessary when a horizontal unit is used.

The speed of brine circulation is especially important in maintaining high heat transfer rate through the evaporator surface. The brine circulator and the tank baffles should be kept in good condition to maintain proper speed of brine circulation.

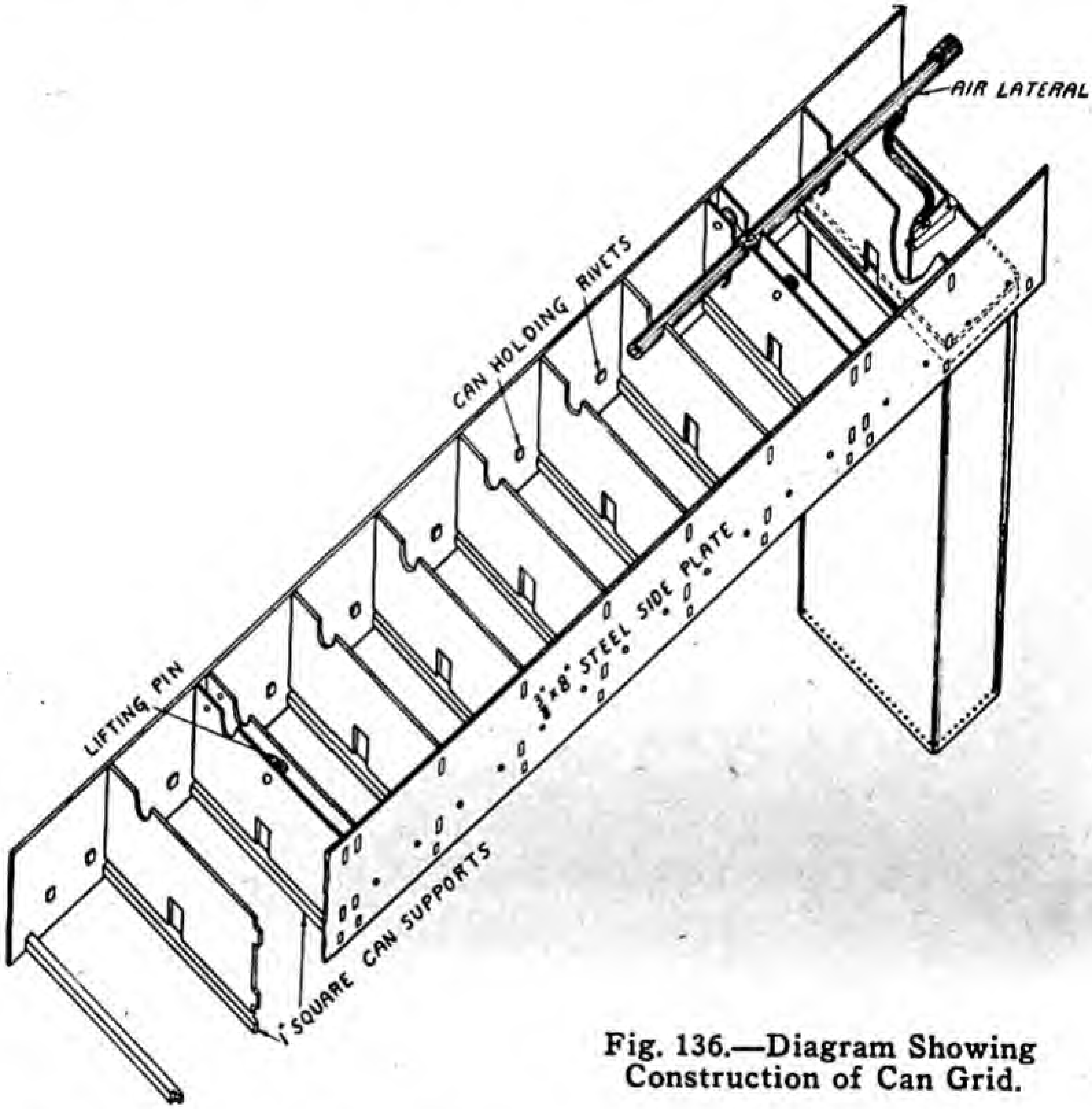


Fig. 136.—Diagram Showing Construction of Can Grid.

Can filling Tanks.—Can filling tanks are commonly used in connection with multiple can hoisting and dumping equipment. Can filling tanks are small compartments which hold enough water to fill one ice can and are generally automatically filled with the proper amount of water for each can. The construction of one type of can filling tank is shown by Fig. 135. The can filling tanks are located at the end of the ice tank and at such an elevation that the water may be drained directly into the ice cans as they are standing upright on the dump. The can filling tanks shown are of the closed cylindrical type. They

should be galvanized to prevent corrosion and consequent contamination of the water.

Can Grids and Baskets.—When multiple can lifting and dumping equipment is used, it is advisable to use can grids or baskets for the purpose of holding a number of the ice cans rigidly together. This facilitates handling of the cans. The constructon of a can grid is illustrated in Fig. 136. The cans are held firmly together between two steel side plates. Provision is made also for the use of an air lateral. Can grids are usually made of black steel and then galvanized or rust-proofed after fabrication. They are made in sizes suitable for handling any number of cans, up to a whole row, across the tank.

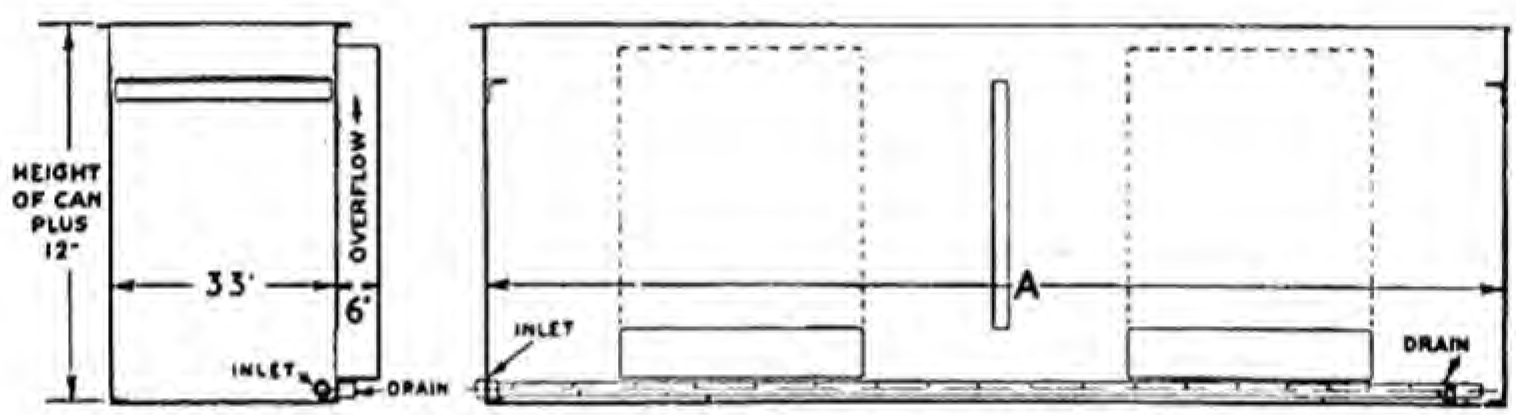


Fig. 137.—Diagram of Thawing Tank.

Can baskets and grids should have sufficient weight so that they will keep the cans submerged to a proper depth. The brine should always be 1-in. to 1½-in. above the finished ice level in the can.

The following tabulation gives some usual specifications of can grids:

TABLE 75.—SPECIFICATIONS OF CAN GRIDS.

Number of Cans	Side Plates	Cross Braces	Lifting Points
4	⅜ in. or ½ in. x 6 in.	½ in. x 6 in.	2
5	⅜ in. or ½ in. x 6 in.	½ in. x 6 in.	2
6	⅜ in. or ½ in. x 6 in.	½ in. x 6 in.	2
8-15	⅜ in. or ½ in. x 6 in.	½ in. x 6 in.	2
16-20	⅜ in. or ½ in. x 6 in.	½ in. x 6 in.	3
21-36	⅜ in. or ½ in. x 6 in.	½ in. x 6 in.	4

For No. 14 gauge cans, the can grids should weigh about 40 lbs. per can. For No. 16 gauge cans, the can grids should weigh about 50 lbs. per can.

Thawing Tanks.—Thawing tanks are usually made of steel, although concrete thawing tanks are used. The steel tanks are commonly made of $\frac{3}{16}$-in. to ¼-in. steel plate, either riveted or welded together. Thawing tanks for five cans or more are generally equipped

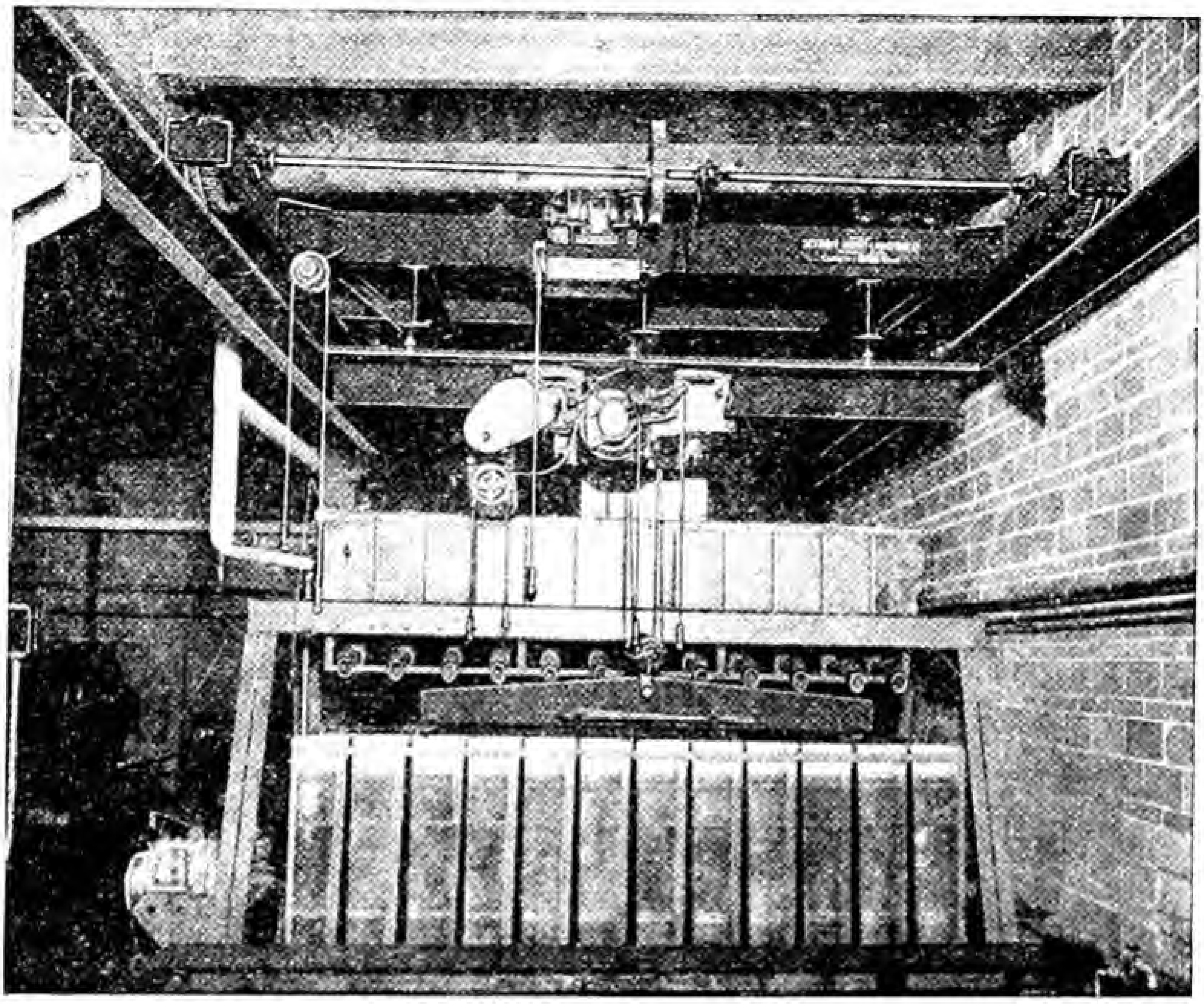


Fig. 138.—Multiple Lift and Hoist.

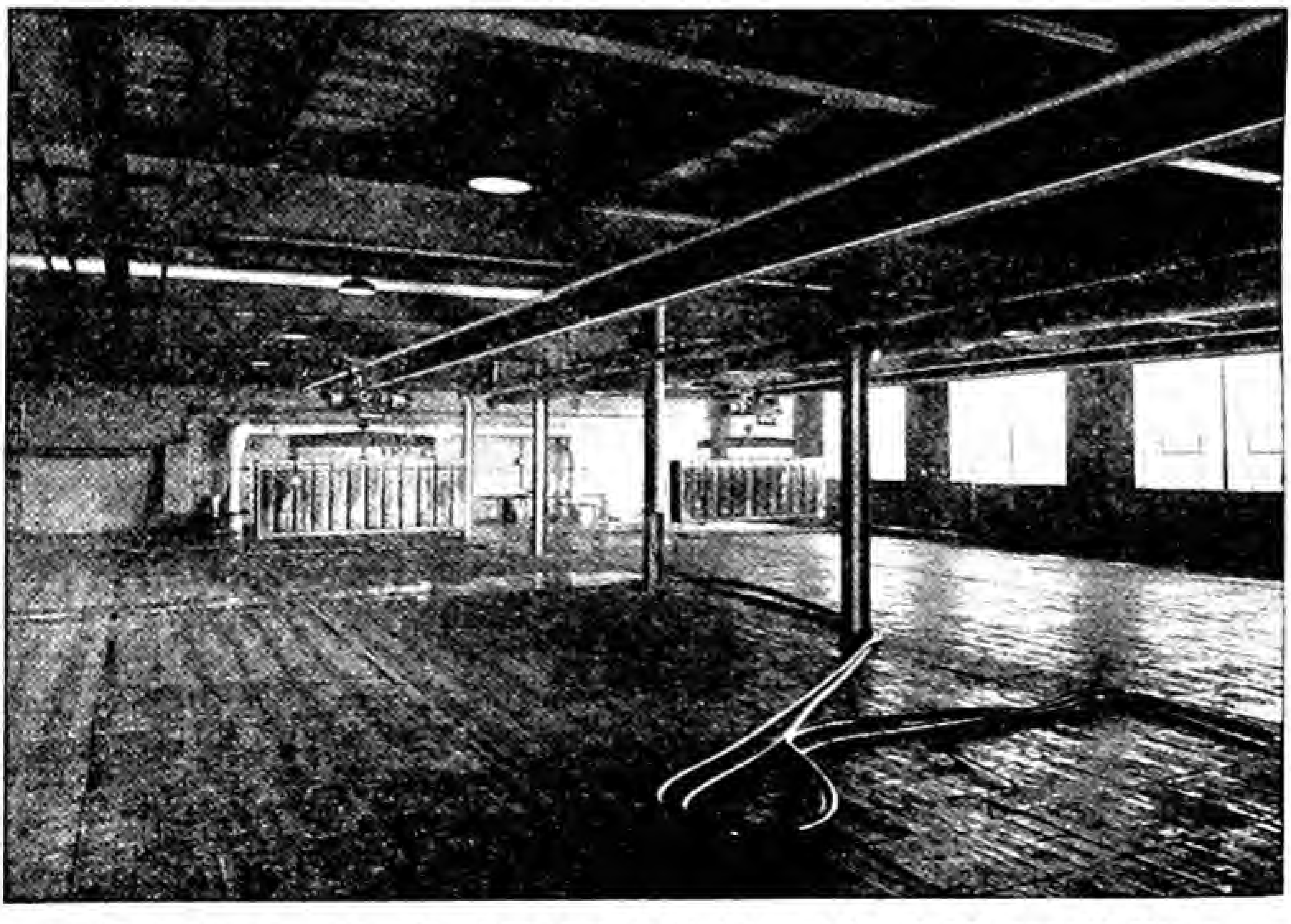


Fig. 139.—Ice Tank Showing Multiple Lift of Monorail Type.

with perforater feed pipes which provide an even distribution of quantity as well as temperature of thawing water throughout the length of the tank. Air agitation is also used sometimes to further aid the even distribution of thawing water. These conditions tend to produce an even thawing of the ice so that all of the blocks will leave the cans at the same time when it is dumped. Fig. 137 illustrates the layout of a steel thawing tank which is used in connection with multiple can lifts.

Water Heaters.—In many ice plants a water heater is placed in the ammonia discharge line for the purpose of obtaining hot water for use in the thawing tank, air header, and thawing needles. These water heaters are usually of the multi-pass shell-and-tube type. In addition to supplying hot water, these heaters also assist in the separation of oil from the ammonia gas as it passes through the heaters.

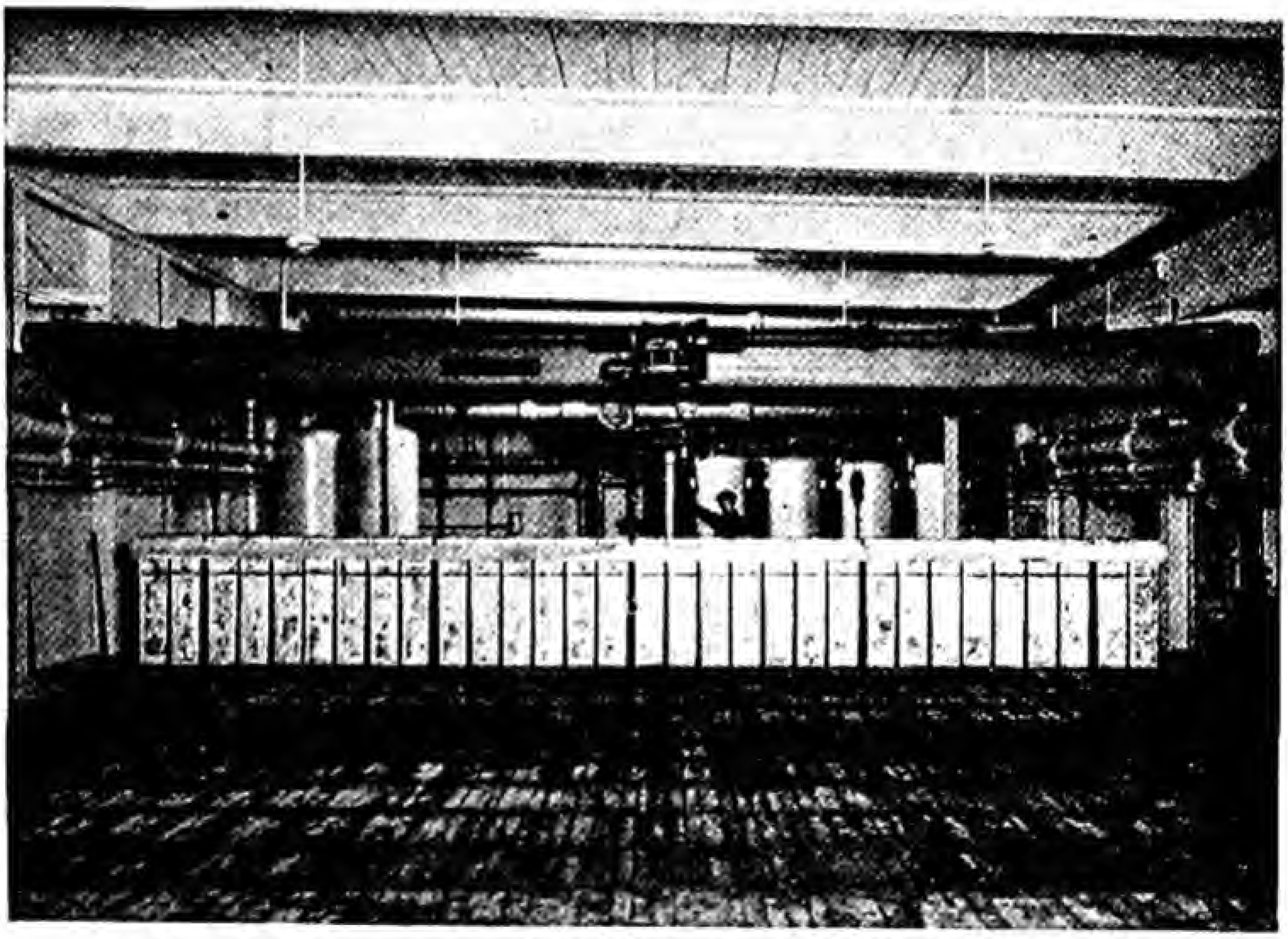


Fig. 140.—Multiple Can Crane and Hoist.

Can Lifting Equipment.—In the larger plants (25 tons and up), the ice cans are commonly hoisted or lifted out of the ice tanks by hoists which handle one-fourth, one-half, or a whole row of ice cans. Various arrangements of cranes, hoists, can fillers, etc., are shown in Figs. 138 to 141.

In Fig. 138, twelve ice cans are handled at one time by means of a bridge type crane.

In Fig. 139, twelve ice cans are hoisted and handled by means of a monorail crane system. In this system, monorail cranes are used to hoist the ice out of the tank and to carry same to end of tank, where it is transferred for the thawing and dumping equipment by means of a transfer crane.

In Fig. 140, 31 400-lb. cans are hoisted and handled at one time. The crane is of the 2-motor, double I-beam type with 5 lifting points.

In Fig. 141 the same crane is shown in the dumping position.

Ice Scoring Machines—After the ice has been harvested in many of the modern ice plants, it is "scored" into 25, 50, or 100-lb. blocks by means of special scoring machines. These machines cut grooves on the blocks of ice by means of revolving saws, so that the blocks will

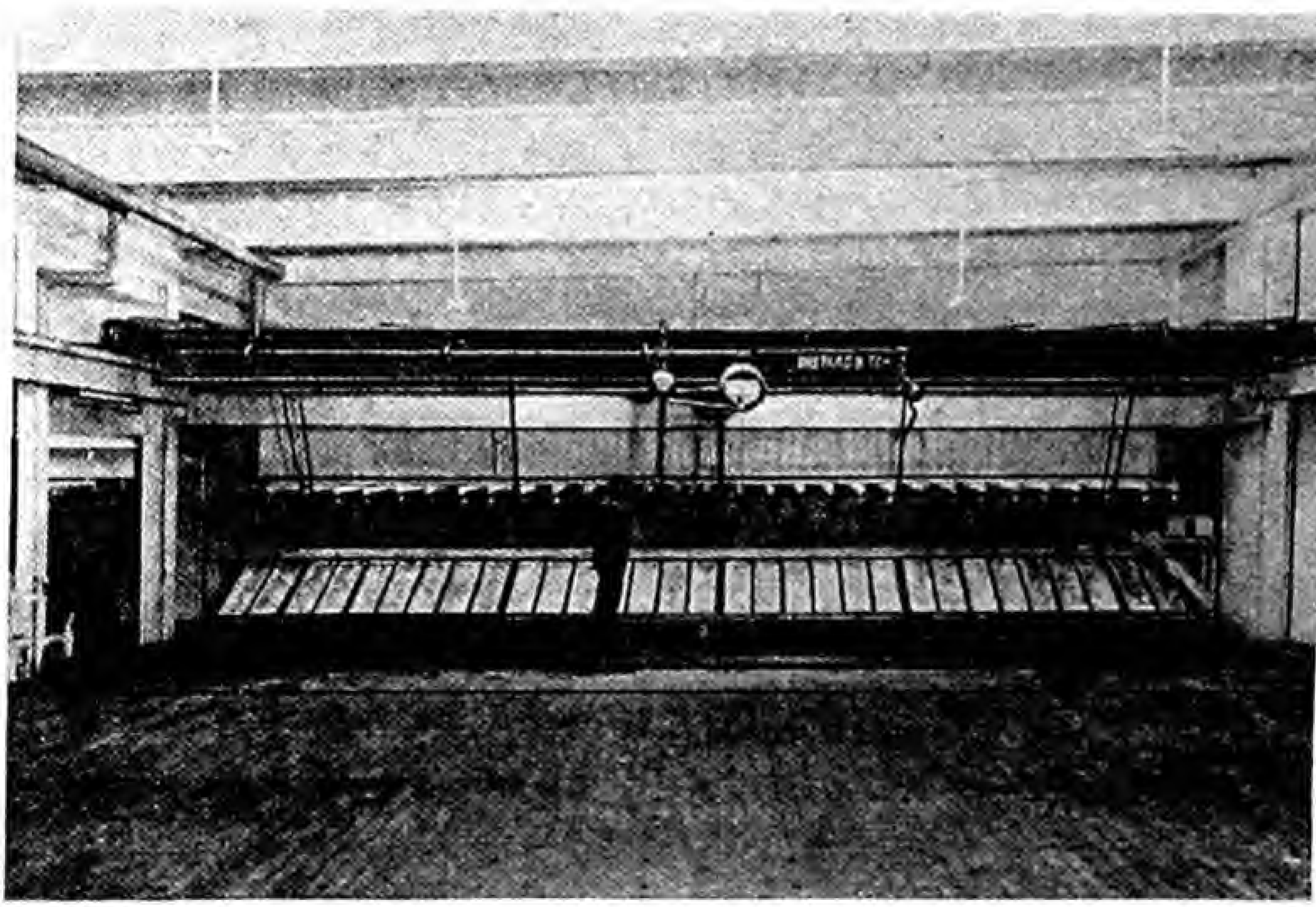


Fig. 141.—Multiple Can Hoist in Dumping Position.

be scored or marked off in the size of pieces wanted.

Some of the advantages of scored ice may be tabulated as follows:

1. It eliminates short weight.
2. It reduces delivery costs.
3. It eliminates the necessity for trained wagon men, because the blocks are accurately marked and are easily separated. This is of special advantage if the ice has a tendency to shatter or check when cut with an ice pick.

QUESTIONS ON CHAPTER XI.

1. Explain the principles underlying the operation of a system for making ice in which a liquefiable fluid is used as a refrigerant.

2. What is the ice making capacity of a 16-in. diameter by 24-in. stroke two-cylinder, vertical, single-acting slow-speed ammonia compressor operating between the gauge pressures of 20 and 125 lbs., with water for making the ice at 65° F.?

3. What is the brake horsepower of the compressor described in Problem 2?

4. In an ammonia compression ice plant, the suction pressure is 20 lbs. and the brine temperature is 10° F. above the temperature of the ammonia. What is the freezing time for standard 300-lb. ice cans?

5. How many 300 lb. ice cans must be installed in an ice tank to produce 50 tons of ice per day under the conditions prescribed in Problem 4?

6. If the coefficient of heat transfer between the brine and ammonia in Problems 4 and 5 is equal to 50 Btu. per hour per sq. ft. per deg. of temp. diff., how many feet of 1¼-in. pipe must be installed in order to produce 50 tons of ice per day under the conditions listed in Problems 4 and 5, if the temperature of the water going to the ice cans is 65° F.?

7. How much shell-and-tube brine cooler surface would be necessary under the same conditions listed in Problem 6, if the coefficient of heat transfer is 80 Btu.?

8. An ice tank makes 60 tons of ice in 400-lb. ice cans per day and has 18,000 ft. of 1¼-inch. direct-expansion pipe. Find the heat transfer coefficient of the coil surface, when the suction pressure is 26 lbs. gauge and water at 75° F. is put into the cans for making ice.

9. An ammonia compression system, operating between the temperature of 5° F. and 86° F. in the evaporator and condenser, is used to manufacture 50 tons of ice per day, with water at 65° F.; find the flooded pipe surface to maintain the brine at 15° F. and the total number of 300-lb. ice cans required.

10. Find the horsepower required to produce a ton of ice under conditions in Problem 9, when vertical single-acting ammonia compressors are used.

CHAPTER XII.

PRODUCTION OF CLEAR ICE.

Ice Freezing Systems.—Having observed those parts of ice manufacturing systems which are more or less common to each system, it is now opportune to study those parts of the various systems which are directly related to the freezing process. The different systems for ice manufacture are characterized by the methods of producing the water for ice making or by the method of freezing the water. Thus, in the distilled water ice making system, water for ice making purposes is secured by condensing the exhaust steam from the steam engine that drives the compressor; while in the raw water ice making system, raw water is frozen into clear ice by agitating the water by some suitable means.

Water for Making Ice.—Generally speaking, any water that is suitable for drinking may be used for making ice. The exact nature of the water determines whether or not the cakes of ice, after they have been frozen, will be clear or opaque in appearance. Water which contains air or appreciable amounts of mineral matter will have an opaque appearance when frozen if means are not taken to prevent this. Water that has been distilled, of course, will be clear after it is frozen, if air is excluded.

When ice was first made, many years ago, distilled water was used entirely. This was due to the fact that the prime movers consisted of steam engines, and it was a simple matter to take the exhaust steam from the steam engine, eliminate the oil, and then condense the steam for the purpose of using it in the ice cans. As the industry advanced, other means of driving the compressors were introduced, such as the electric motor, the oil and gas engine, etc. In this type of plant it was necessary to either provide means of producing distilled water or provide means of producing clear ice from water which had not been distilled, or which is commonly termed "raw water." Water taken from rivers, lakes, wells, etc., is, therefore, termed "raw water" and the modern plant uses such raw water and makes clear ice.

It is well to note the exact process of freezing when the water contains air, mineral matter, or suspended solids. As soon as the water reaches the freezing point, crystals of pure ice begin to form. These crystals are small and contain only pure water. Small amounts of air, mineral matter and suspended solids are caught and held between the pure ice crystals during the process of freezing. The formation of ice in this manner gives it the opaque appearance.

Many means have been devised for keeping the foreign matter from freezing into the blocks of ice. In some cases the ice cans may be moved or tilted in order to accomplish this. In other cases the water may be circulated from one can to another. The method that is commonly used at present consists of agitating the water in the cans by means of compressed air. The air pressure to be used for any particular plant depends upon the nature of the water and the local conditions in the plant, so that the present type of air agitated systems may be divided into the low-pressure and the high-pressure type. In the low-pressure type, the pressure of the air used for agitating the water will vary from 2 to 5 lbs., while in the high-pressure system it will vary from 15 to 30 lbs. In all of these cases the primary object is to produce clear ice.

As far as the refrigerating power is concerned, clear ice and opaque ice have nearly the same cooling effect. From a pathological viewpoint, it may be said that clear ice and opaque ice are equally pure. The purity in any case, of course, would depend upon the water supply. It seems that the general public has gotten into the habit of demanding clear ice, which accounts for its manufacture at present.

Low-Pressure Air System.—As previously indicated, in this type of system, air at a pressure of 2 to 5 lbs. is used for agitating the water. The usual arrangement of this system may be ascertained by inspecting Fig. 142. The air is conducted to the cans through a system of air headers and lateral pipes. Drop pipes which extend to within 8 to 12 in. of the bottom of the can are connected to an automatic air valve which is located in the lateral pipes. By placing the drop-pipes with their connections on the automatic air valve in the lateral pipe, a supply of air at low pressure is admitted to the can.

The introduction of air into the bottom of the can in this manner causes the water to be agitated in a manner similar to boiling. The movement of the water across the freezing surface of the ice prevents air and other matter from becoming enclosed with the freezing ice crystals.

The general arrangement of the whole system is shown by Fig. 133. The apparatus consists generally of a motor for driving the rotary low-pressure air blower (centrifugal blowers are used also) which

supplies the air to the pipe system on the tank top. In some cases an air conditioner for cooling and purifying the air is used between the blower and the tank top piping. The function of the rotary air blower is simply to compress the air to a pressure of 2 to 5 lbs. so that it will

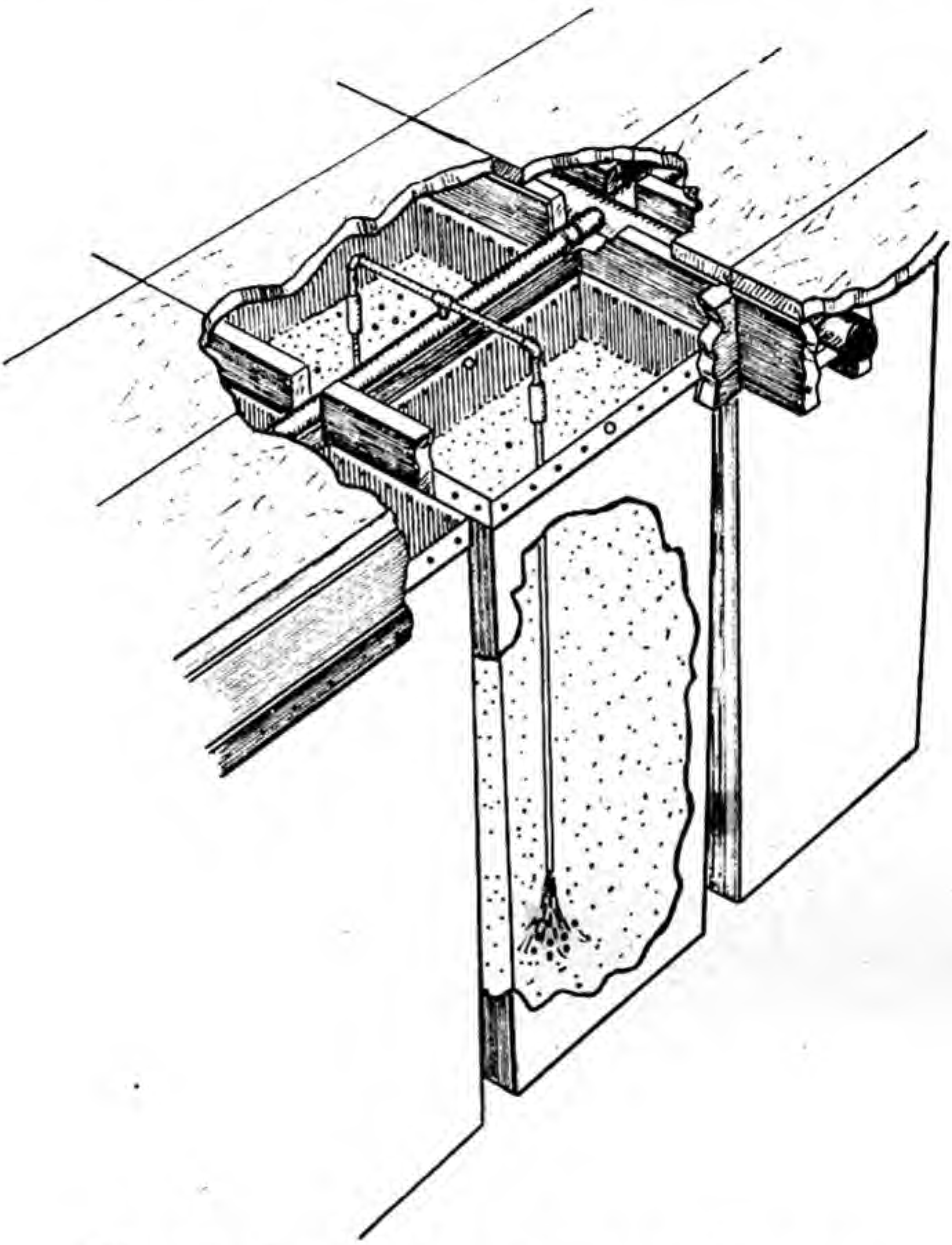


Fig. 142.—Low-Pressure Air Agitating System.

flow through the piping system into the water at the bottom of the cans. The air conditioner may be cooled either by means of cold water or brine from the ice tank. It serves the purpose of cooling the air as well as removing part of the moisture. The presence of moisture in the air circulating system may, of course, lead to operating difficulties in the event that the water becomes frozen in the connections. This is not very probable with the arrangement shown in Fig. 133, except

in the winter months, since no part of the connections comes in contact with the cold brine. In colder climates it is sometimes advisable to place hot water lines in the main air headers to heat the air so that moisture will not condense in the headers.

The main air headers extend from one side of the tank to the other at one or more convenient places along the length of the tank. The lateral pipes extend between the headers and between each alternate row of cans. All piping and fittings are either galvanized or made of brass.

The air continues to bubble up through the water until the block of ice is frozen nearly solid. At this point, due to the concentration of the impurities in the core, it is sometimes necessary to pump out this core and refill the space with water from the storage tank. The cold water is generally removed by means of a small motor-driven centrifugal pump which is connected by means of a flexible hose to a suitable core suction nozzle, similar to the arrangement shown in Fig. 133.

After the core space has been refilled with fresh water, the drop-pipes are permitted to remain in the cans until the ice is very near the end of the pipe. The drop-pipes are then removed and placed upon a convenient rack. The automatic air valve in the lateral stops the flow of air as soon as the drop-pipes with their connections are removed from the cans. As soon as the water in the core space has entirely solidified, the cans with the ice may be removed from the tank in the usual manner.

This type of system is well adapted for the freezing of relatively pure water and for use in the small and medium sized plants. The initial cost and the power cost are low. However, the cost of labor for harvesting the ice may become quite large. This is due to the fact that individual attention must be given to each of the double drop-pipes. To eliminate part of this extra labor several drop-pipes are sometimes connected to an air lateral, in which case the system is called "the multiple drop-tube type." This is shown by Fig. 143. In order to freeze ice with a smaller core many plants use drop tubes with side perforations which freeze in the block and then are thawed out with a warm water needle.

High-Pressure Air System.—In the high-pressure air system, air is compressed to 20 to 30 pounds and then expanded to a working pressure of 15 to 18 lbs. on the air lateral. The arrangements of the air pipes in the cans are somewhat different from that of the low-pressure system. In the high-pressure system, the air tubes in the cans may be soldered in the corner of the can, soldered on the side of the can, fastened on the outside of the can and brought into the can at the bottom, or they may be suspended from the end of the can at the top.

From this, it will be noted that the air tubes in one case are fastened permanently to the ice can, while in the other case they are suspended from the end of the can. In all cases, the air pipe in the can is allowed to remain in its place until the block of ice has entirely solidified. In the event that the air pipe is allowed to freeze solidly into the block of ice, it, of course, must be removed by means of thawing.

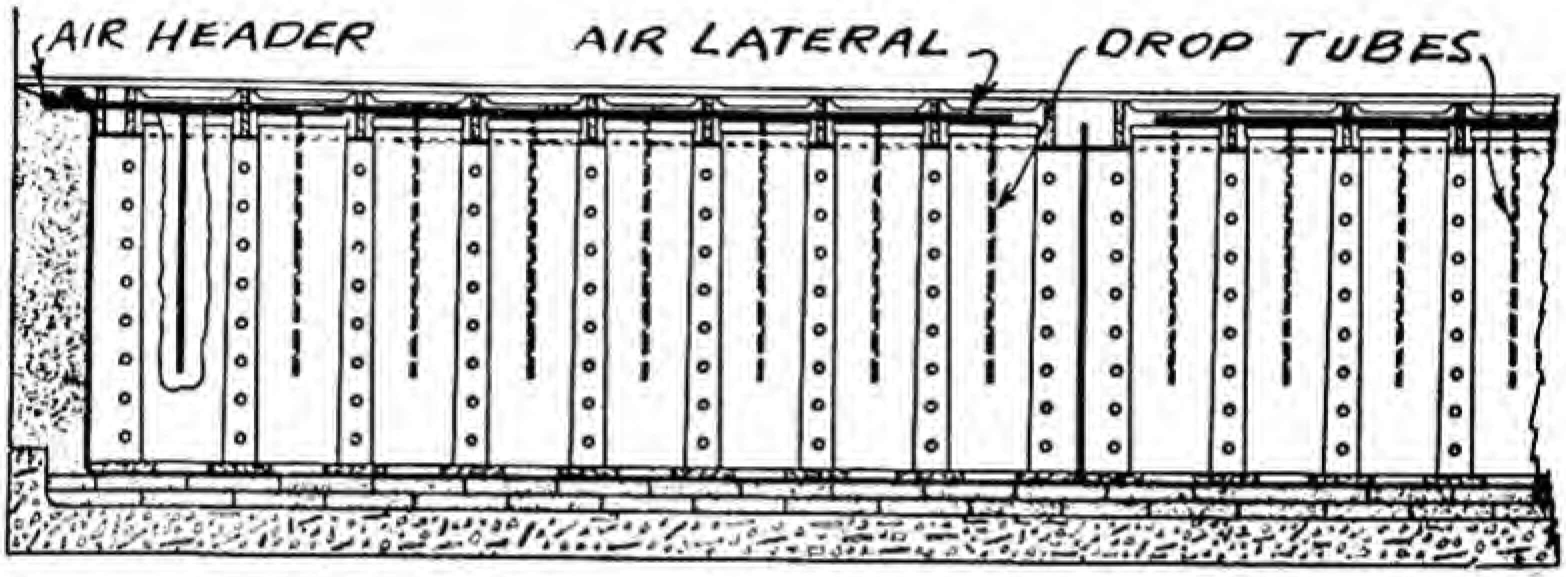


Fig. 143.—Multiple Drop-Tube System.

The general arrangement of the high-pressure air system is illustrated by Fig. 144. The apparatus generally consists of a piston compressor for compressing the air to a pressure not exceeding 30 pounds. The air, after being discharged through the compressor, is led through a water-cooled precooler for the purpose of reducing the temperature of the air. From the precooler the air is taken to an air conditioner which is generally cooled by brine. The function of the air conditioner is not only to further lower the temperature, but also to reduce the moisture content. This is due to the fact that the moisture must be eliminated as nearly as possible from the air so that it will not become frozen in the various air connections. This is especially true when the air tube on the ice can is on the outside of the can, in which case the air tube is subjected to the temperature of the brine. Since the moisture condensed out of the air will collect and freeze upon the cold brine coils in the air conditioner, this apparatus is generally made in duplicate units, so that one will cool the air while the other is being defrosted.

In case that the water contains a small amount of mineral matter, the block may be allowed to freeze solid without removing the core. In some cases, when the water contains exceptionally large amounts of mineral matter, it is necessary to remove the core in the usual manner as indicated in the low-pressure system. Of course, the ice may be harvested in the usual manner.

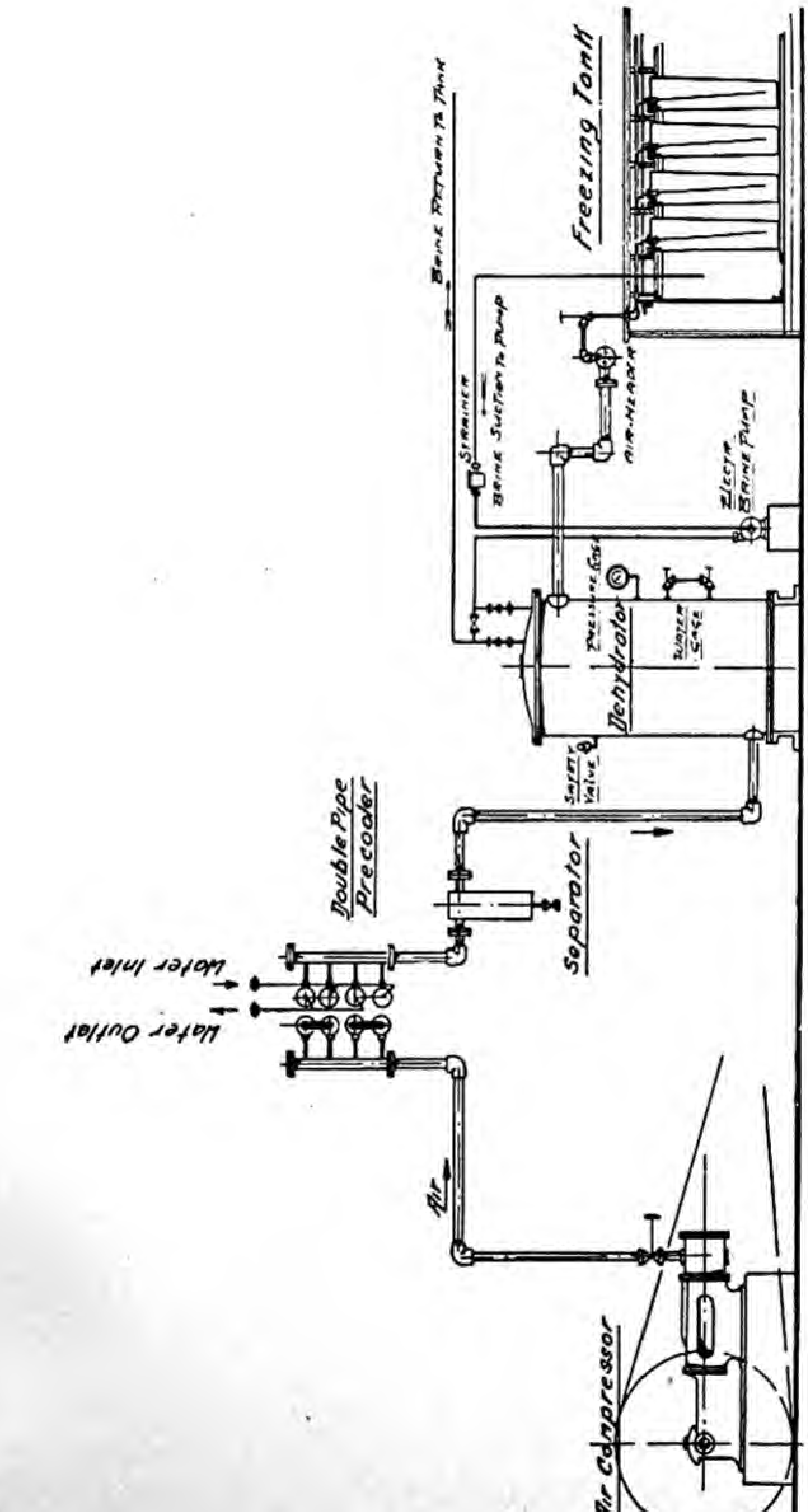


Fig.144.—High-Pressure Air Agitating System.

This type of plant is particularly well adapted for freezing waters which contain excessive amounts of mineral matter. Under general plant conditions, this type of air agitating system does not require as much attention as does the low-pressure system. These facts make the system well adapted for use on all sizes of plants, including the very largest.

The construction of single-cylinder air compressors is illustrated in Fig. 145.

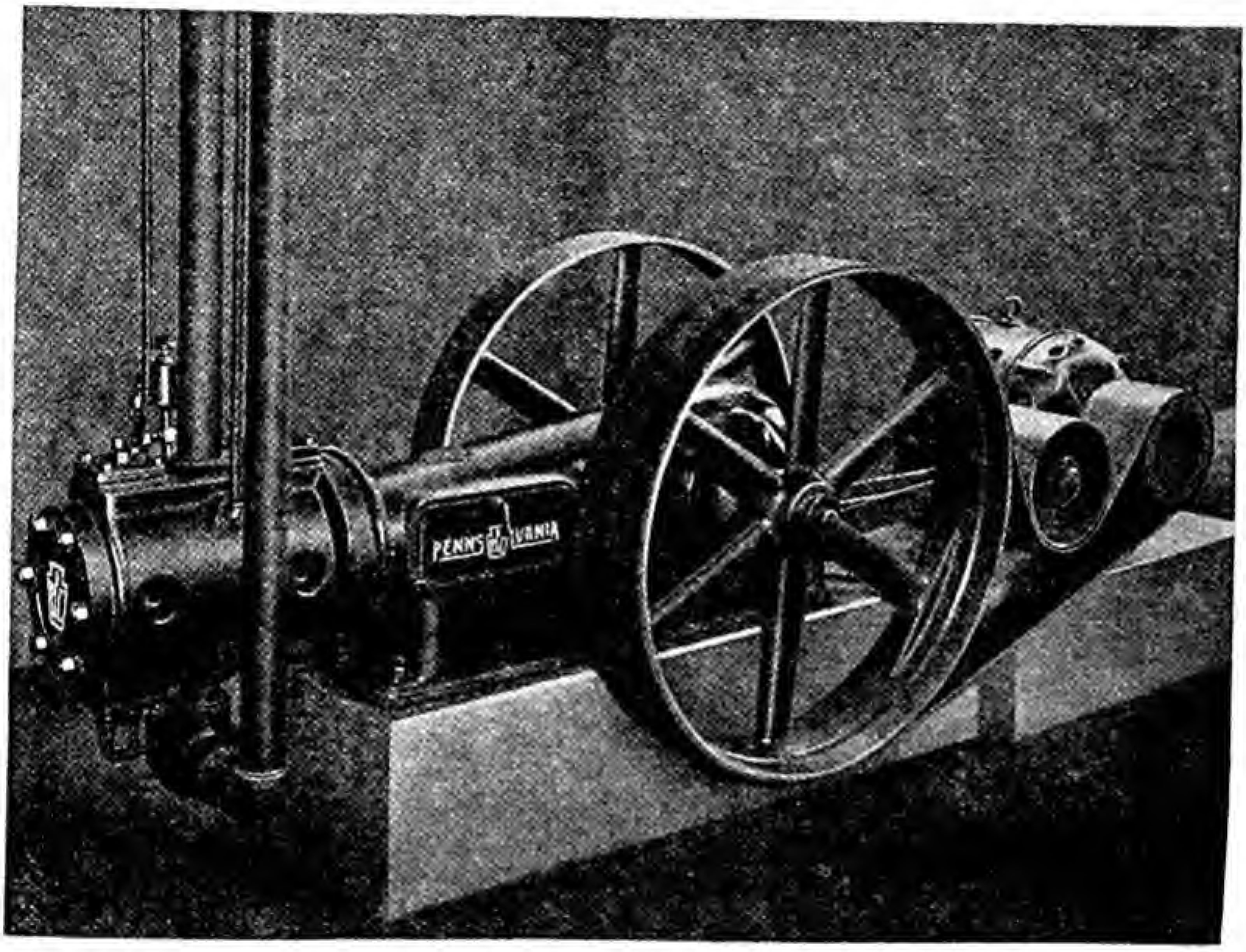


Fig. 145.—Single-Cylinder Air Compressor.

Water Purification.—Various kinds of waters which are available for ice making are seldom suitable for producing clear ice without some means of reducing mineral content. In general, water will contain dissolved mineral matter and suspended matter. The three general methods of purification usually employed, consist of chemical treatment, distillation, and the base exchange process. Of course, in the case that the water is relatively pure, it is not necessary to provide treatment. The effects of dissolved minerals are the formation of sediment, discoloration, or deposit when the water is frozen into a block of ice. The effects of the individual minerals usually found in water may be ascertained by an inspection of Table 76.

TABLE 76.—EFFECTS OF FOREIGN MATTER IN WATER FOR MAKING ICE.

Minerals in Water and their Symbols	Effect in Ice	Result of Treatment with Hydrated Lime
*Calcium Carbonate ($CaCO_3$) or Calcium Bi-carbonate ($CaCO_3 \cdot H_2CO_3$)	Form gritty, dirty colored deposit in the ice, usually in lower part and center of cakes. Cause shattering at low freezing temperatures.	Elimination
*Magnesium Carbonate ($MgCO_3$) or Magnesium Bi-carbonate ($MgCO_3 \cdot H_2CO_3$)	Form gritty, dirty colored deposit in the ice, give unclear appearance and also cause shattering at low freezing temperatures.	Elimination
*Magnesium Sulphate ($Mg\ SO_4$) †Magnesium Chloride ($Mg\ Cl_2$)	Form opaque greenish or grayish colored cakes, concentrate mostly in cores and retard freezing. Often show up as white ice and dirty colored streaks, spots and heavy cores.	Changes to Calcium Sulphate Changes to Calcium Chloride
Sodium Carbonate ($Na_2\ CO_3$) or Sodium Bi-carbonate ($Na_2CO_3 \cdot H_2CO_3$)	In only small quantities these carbonates are known to be a cause for shattering at freezing temperatures less than 15°, also	Purification improves but little; water with much of these carbonates present is unsuitable for ice making. Distillation is the only way to remove sodium salts.
Sodium Sulphate ($Na_2\ SO_4$) Sodium Chloride ($Na\ Cl$)	Cause white ice, concentrate in the cores and retard freezing, but do not form deposit. Also cause white shell around bottoms of cakes.	
*Calcium Sulphate ($CaSO_4$) †Calcium Chloride ($CaCl_2$)	Act like and are no worse than the same salts of sodium, but are much less objectionable than the same salts of magnesium.	Soda ash changes to sodium sulphate. Soda ash changes to sodium chloride. These changes with soda ash are seldom necessary for ice making but always are for boiler feed.
*Iron Oxide—Rust ($Fe_2\ O_3$)	Causes bad discoloration, yellow or brown deposit.	Elimination.
*Aluminum Oxide ($Al_2\ O_3$)	Causes dirty deposit.	Elimination.
*Silica (Sand) (SiO_2) Suspended Matter	Cause sediment.	Elimination.
Organic Matter	Causes discolored cores.	Elimination.

*These minerals are known as incrusting or scale-forming substances.

†The calcium and magnesium chlorides are classed as corrosives, but all forms of calcium and magnesium are classed as "hardening salts." The sodium salts do not form scale in a boiler, but they cause foaming under certain conditions. They are classed as non-incrusting substances.

The first column of this table gives the names of the common minerals in water, together with their chemical symbols. The second column gives the various effects of such foreign matter in the ice. The third column indicates the result of treatment with hydrated lime. The most common impurities are those of calcium bicarbonate and magnesium bicarbonate. When waters containing these bicarbonates are either boiled or frozen, the bicarbonates are changed into carbonates, and carbon dioxide gas is given off. The carbonates will form a sediment in the block of ice, while the presence of the carbon dioxide gas will give the cake of ice an opaque appearance.

Minerals, in addition to formation of sediments, discolorations, or deposits, tend to cause the ice to shatter and crack when they are present in large quantities. The usual method of treating water for ice-making purposes employs a water softener which uses lime and coagulants. The effect of softening water by means of lime may be ascertained by inspecting the third column in Table 76.

Sulphates, chlorides and nitrates of calcium, magnesium and sodium are generally found in water which is available for making ice. This class of mineral matter does not form scale, but tends to make the water hard. During the process of freezing they do not form a deposit, but merely become concentrated in the core. They may become so concentrated as to produce a discoloration of the ice and to retard the freezing process. The presence of even a small amount of iron oxide causes a bad discoloration in the ice, generally having a yellow or brown color. If the iron oxide amounts to more than 0.02 grains per gallon, it is necessary to remove it by treatment. Suspended matter of any kind, of course, is objectionable and may be removed simply by the proper filtration.

The amount of organic matter present in water will depend upon the locality of the plant, the season of the year, etc. Generally, it is present only during summer months. This is eliminated in the lime softener by coagulation and filtration. The elimination of the organic matter in all cases is necessary, due to the fact that it not only causes discoloration of the core, but such discoloration may extend to all parts of the block of ice.

When a can filled with water containing the foregoing impurities begins to freeze the first shell of ice that is formed is almost entirely free from minerals. Thus, this first ice shell is made of pure water, since it contains very little of the minerals present in the water before freezing. As the freezing process continues, the minerals and suspended matter are moved away from the ice surface by the currents set up in the water by the air agitation. This matter collects in the unfrozen water which finally becomes the core. As the freezing continues the concentration of the mineral matter becomes greater, so

that in the final stages the mineral matter begins to freeze into the ice, causing a deposit, discoloration, or other difficulties. Thus, it will be seen that the mineral matter and any suspended matter are finally concentrated in the core. Some of the difficulties arising from this action may be eliminated by pumping out this core water.

When sulphates and chlorides are present in the water in sufficient quantities, they will tend to freeze into the ice during the early part of the freezing period, causing a white shell to be formed; but, as the process of freezing continues they become concentrated in the cold water, making the same brackish and briny, which retards the freezing.

The carbonates of calcium and magnesium begin to freeze into the ice at a much earlier time than the other mineral matters. As previously indicated, these carbonates form sediment. This insoluble sediment is deposited in the lower and center parts of the cake. When the water contains a large amount of these minerals, this difficulty may be somewhat avoided by pumping out large cores.

The deposits which are formed by the carbonates of calcium and magnesium remain insoluble when the ice melts, so that sediment is left in the ice box or any other place where the ice is melted. The other minerals which are soluble in the water redissolve as the ice melts and are drained away with the water.

The total concentration of minerals in the water is the limiting factor although as mentioned before, certain minerals have particular effects in the freezing of good quality ice. Where sulphates predominate the core is likely to be thick and heavy. This is not so true of chlorides and if the chloride content is equal to 10 percent or more of the sulphates the water will have a chloride type and cores will be lighter and narrower.

On the other hand chlorides produce a greater tendency to shatter and crack the frozen blocks. This tendency is considerably retarded by treatment which introduces ammonium ions into the water. Usually ammonium sulphate or ammonium chloride is used for this purpose. The choice of which one depends upon the type of the water initially —whether the chloride to sulphate ratio is above or below 10 percent. University of Illinois Bulletins Nos. 253 and 254 (by Dana Burks, Jr., 1933) report an exhaustive investigation of water treatment and freezing methods. They are the basis for the brief recommendations given above.

The demineralizing treatment employing lime and alum or lime and sodium aluminate operates as follows: The first stage of the process consists of carefully proportioned and uniform mixing of the chemicals with the raw water. Various mechanical means for producing this result are employed, the primary considerations being a uniform mixing of the chemicals in the water and the maintaing of a correct

proportion between the chemicals and the water. The water, after becoming charged with the chemicals, is led to a settling tank where suspended matter is allowed to settle out in a few hours. The precipitated matter collects in the bottom of the settling tank and may be drained off at suitable intervals. The softened water is then drained off at the top of the settling tank, and after passing through a suitable filter it is conducted to a cooling tank or to the ice cans. The function of the filter is to remove all traces of suspended matter.

A base exchange method of direct removal of any desired proportion of the mineral content of water has been developed recently. It not only removes the calcium and magnesium carbonates but also the sulphates and chlorides as well. This equipment can produce water of a purity equivalent to distillation although for purposes of economy in cost the systems installed in ice plants do not demineralize the water to such a great extent. The cost of the equipment and its operation is well within the economic range of the ice plant.

The direct distillation method was, of course, the method used in the beginning. This consisted of the distillation of the water from the steam boiler, the use of same in the steam engine, and finally the condensation of it in a steam condenser. This process, of course, may be used with water containing any reasonable amount of mineral matter, since this matter is left in the evaporator or boiler. It is evident, however, that many difficulties of operation will occur when water containing excessive amounts of mineral matter is used. The formation of scale in the evaporator or boiler may be practically eliminated by the use of a suitable water softener.

The construction of the water treating equipment of the International Filter Co. is illustrated in Figs. 146 and 147.

They diagrammatically show the standard water treating systems. Under certain conditions and when handling unusual waters, the recommended treating systems may differ slightly from those shown but these drawings indicate the standard arrangement used in the great majority of cases.

Fig. 146 indicates the system where lime is the principal treatment. Fig. 147 indicates the treating plant which provides for lime treatment followed by the Neutralizer treatment.

Referring to Fig. 146: The chemical mixing and feeding tank can be top operated as shown or ground operated as indicated by dotted lines. The automatic water control is always placed on top the sedimentation tank immediately over the downtake. The open cast iron coagulant feeder is placed alongside the water control and automatically feeds a small amount of alum as an aid to sedimentation.

The chemical mixing tank is provided with a series of plow shaped agitators revolving in a vertical plane, thus providing up-and-down

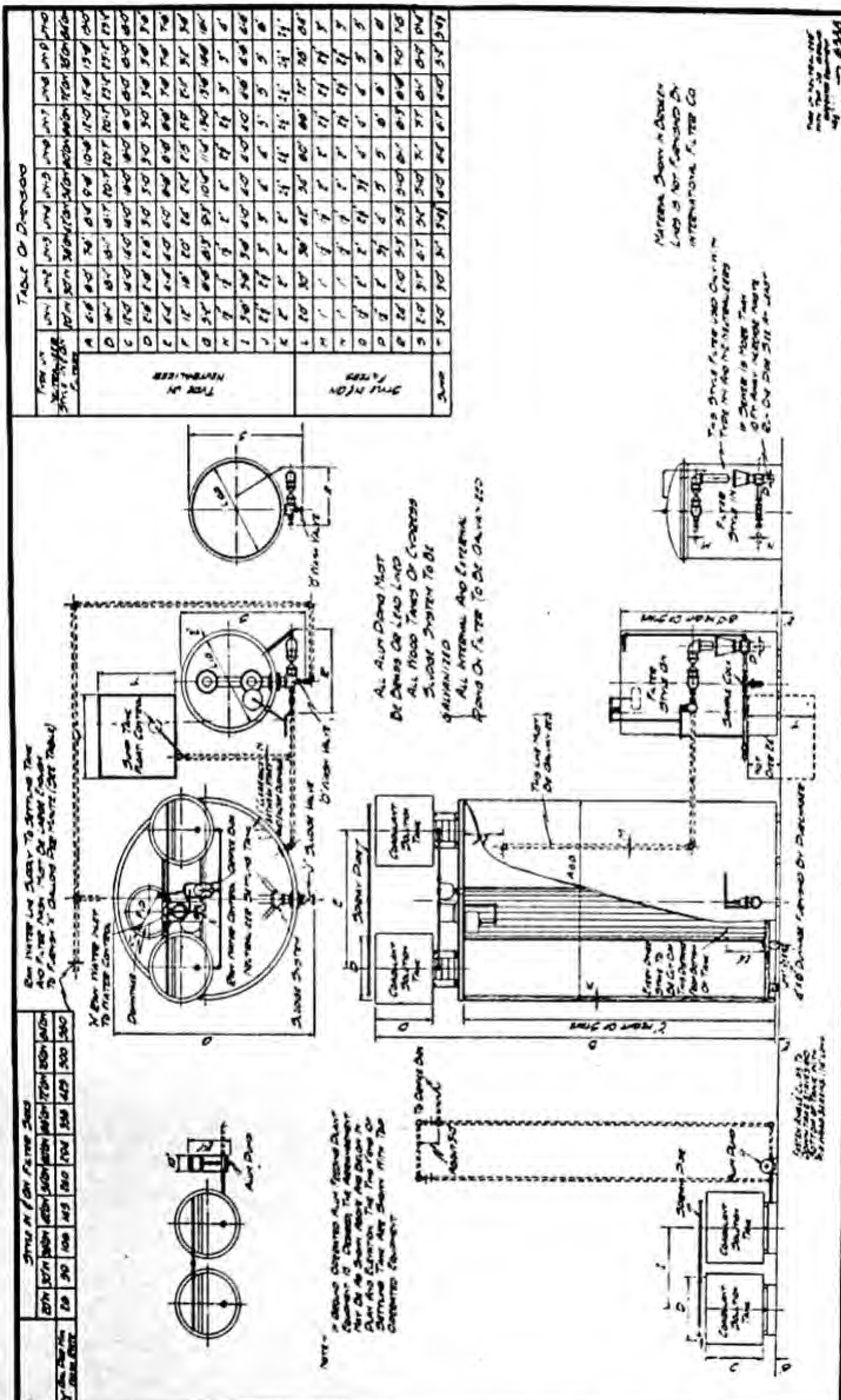


Fig. 146.—Water Softening Plant and Storage Tank.

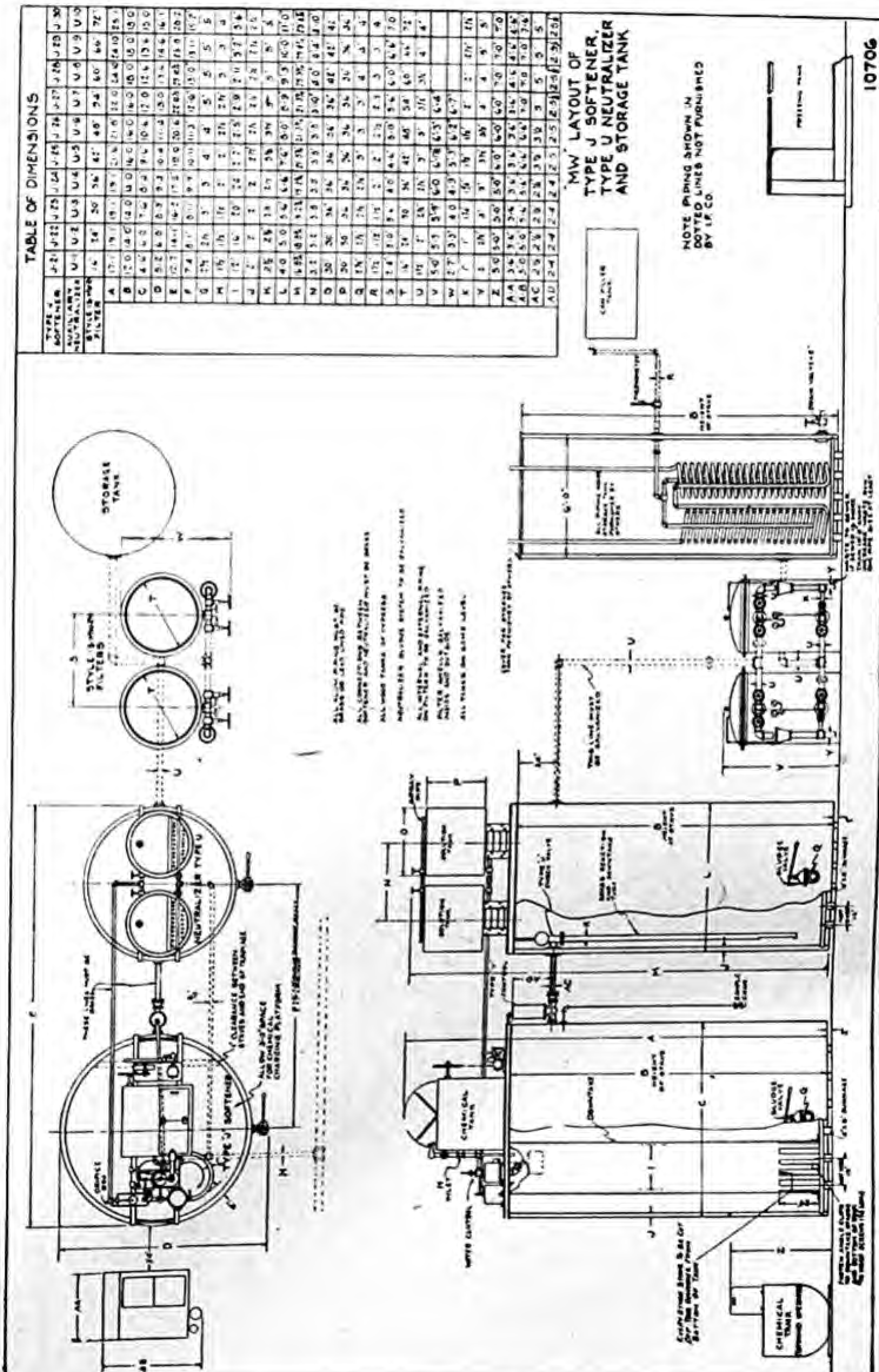


Fig. 147.—Softener, Neutralizer, and Storage Tank.

agitation and maintenance of a uniform strength of chemical mixture. One set of agitators is provided with chemical cups which pick up the chemical mixture and deliver a measured stream into a collector funnel which delivers the solution to the raw water discharging into the downtake or reaction tube.

The automatic water control consists essentially of a standard orifice over which a constant head of water is maintained by a control float and balanced valve. The control is such that when the treating plant is in operation water enters the softener at a constant rate of flow. The chemical mixing and feeding equipment delivers measured chemical at a constant rate of flow thus insuring absolute uniformity of treatment.

The water control is provided with a trip assembly actuated by a float riding on the surface of the water in the sedimentation tank, so that when the amount of treated water used does not equal the capacity of the plant, the trip assembly will automatically stop the motor driving the chemical mixer and shut off the balanced control valve. The water control and chemical feed will then be inoperative until the water level drops a pre-determined distance, at which time the trip assembly will automatically open the water inlet valve and start the electric motor driving the chemical mixer and feeder, automatically putting the plant back in operation.

The measured water and measured chemical meet at the top of the downtake, the stream of water being directed against the side of the downtake so as to impart a rotary motion to the water and thus provide for proper mixing and the beginning of efficient sedimentation.

At the bottom of the downtake the treated water with precipitated impurities passes out through the slotted openings and with a continued, but much slower rotary motion, the water passes upward, the sludge settling back through the water to the bottom of the tank where it is periodically blown off through the sludge system.

The settled water is taken off near the top and passes to the twin sand filters. Entering the filters, the treated water passes downward through a bed of filter sand, then through layers of graded filter gravel and out through the collector systems. From the filters the water then flows by gravity to the storage tank.

When the filters become clogged they are back washed by a reverse flow of filtered water from the storage tank, the inlet valve being closed and the waste valve opened. The wash water is distributed by the manifold system, further distributed by the graded gravel and then passes upward through the sand bed, putting the latter in suspension and washing the collected impurities out to waste.

The storage tank and sedimentation tank are of the same height so that the water level will equalize in the two if the filters are clean.

The filters are of such size that both of them, operating in parallel, will have a normal rate of filtration of but 1½ g.p.m. per sq. ft. of filter area, based on the rated capacity of the treating plant.

The chilled water is drawn from the storage tank to the can filler tank and from there placed in the ice cans which are then lowered into the freezing tank.

Referring to Figure 147: The arrangement of this type of treating plant is very similar to that shown in Figure 146 except that the neutralizer treating equipment is placed between the lime treater and the filters.

The function of the neutralizer is to convert the deposit forming mineral matter that remains in the water after lime treatment to a more soluble salt which is not objectionable in these respects.

When natural waters are properly treated with lime the objectionable carbonates of calcium and magnesium, the iron oxide, the silica and suspended matter are almost completely removed and consequently the difficulties they cause are very largely overcome.

Complete elimination of all of these objectionable substances is not obtained, however, because the calcium carbonate is slightly soluble in water and there will remain, even after accurate lime treatment, two or three grains of this soluble calcium carbonate. When the water is frozen, this calcium carbonate will form a slight deposit in the core of the ice, leaving a slime after the ice has melted.

By treating the water after lime treatment, in the neutralizer, the small amount of calçium carbonate remaining is changed to the soluble calcium sulphate. This eliminates the possibility of traces of deposit and also reduces the tendency to cause checking and cracking. As a result of lime treatment followed by neutralizer treatment, ice entirely free from deposit may be produced and at the same time the water may be frozen at temperatures appreciably lower than is possible with untreated water.

Using the combination lime and neutralizer treatment (see Figure 147) the arrangement of the lime treater, filters, storage tank and can filler tank are similar to that described under Figure 146 with the neutralizer placed between the lime treater and filters. The equipment can be arranged for either top or ground operation.

Wood solution tanks are provided for making up the neutralizing solution (iron free aluminum sulphate). These tanks are connected with the orifice measuring box—placed alongside and operated by the water control trip assembly on the lime treater. Whenever water enters the first treating tank a proportionate amount of neutralizing solution is fed from the orifice box into the type U control reservoir.

The neutralizer control reservoir is placed between the lime treater and neutralizer. The diameter of the reservoir chamber is in definite

proportion to the diameter of the first settling tank, and of such size that each inch of depth will provide sufficient solution to treat an amount of water equivalent to each inch of depth in the first settling tank.

Therefore, whether or not water is entering the lime treater, the level in the reservoir will change as the level in the first settling tank changes and the proper amount of solution will be fed to the water passing to the neutralizer tank.

The neutralizing reaction is practically instantaneous, taking place in the brass reaction or downtake tube in the neutralizer tank. Two hours holding time is provided in the latter so as to give opportunity for ample coagulation and sedimentation before the water passes to the twin filters.

Air for Agitation.—The amount of air that is required for properly agitating the water in the cans during the freezing process seems to depend in general upon two factors—speed of freezing and the relative amount of mineral matter present in the water. As previously indicated, air and carbon dioxide are present in the water during the freezing process, so that the volume of air circulated through the water must be increased as the speed of the freezing is increased to prevent these gases from being frozen into ice.

In general, it may be said that the volume of air admitted to the ice cans should be increased from 1 to 2½ per cent for each degree of temperature that the brine is below 14° F. The amount of air to be admitted to the cans must be increased as the amount of mineral matter present is increased. The amount of mineral matter in the water seems to be the principal determining factor in the consideration of the amount of the air agitation. Since the outer part of the block near the can freezes faster, a number of plants using water with higher mineral content employ a greater volume of air when the can is placed in the brine and then reduce the air flow when 1 in. or more of ice has frozen in the can.

It seems that the pressure required for an air agitating system is of secondary importance, and that the circulation of the proper volume of air through the water is of primary importance. In either the high or low pressure system, during the principal part of the freezing process, it is evident that the pressure of the air at the bottom of the can will depend upon the depth of the water in the can. The use of high pressure is, therefore, employed to give a large volume of air and to make it possible to continue the agitation until the cake of ice has entirely solidified.

The air pressure required for agitating water in the different sizes of ice cans will vary from 2 to 5 lbs. in the low-pressure system, 2 lbs.

TABLE 77.—VOLUME OF AIR AND HORSEPOWER REQUIRED FOR VARIOUS TYPES OF CLEAR RAW WATER ICE SYSTEMS AT USUAL OPERATING PRESSURES.

Jos. A. Martocello.

Number of Cans	LOW PRESSURE PERFORATED "FREEZE-IN" DROP TUBES							Low Pressure Old Style ½ Cu. Ft. Air Per Can Per Min.		High Pressure "Freeze-in" Drop Tubes 1/7 Cu. Ft. Air Per Can Per Min.		High Pressure Stationary Tubes 1/5 Cu. Ft. Air Per Can Per Min.	
	Cu. Ft. Per Can Per Min.	Total Cu. Ft. Per Min.	1 Lb.	1¼ Lbs.	1½ Lbs.	1¾ Lbs.	2 Lbs.	3 Lbs. Press.		18 Lbs. Press.		20 Lbs. Press.	
			Brake H.P.	Brake H.P.	Brake H.P.	Brake H.P.	Brake H.P.	Cu. Ft. Per Min.	Brake H.P.	Cu. Ft. Per Min.	Brake H.P.	Cu. Ft. Per Min.	Brake H.P.
50	5/8	31.5	.18	.26	.35	.42	.47	25	.7	7	.63	10	.95
100	5/8	63.0	.35	.50	.70	.80	.80	50	1.2	14	1.26	20	1.90
150	5/8	94.0	.51	74	.90	1.10	1.20	75	1.7	21	1.89	30	2.85
200	5/8	125.0	67	.98	1.10	1.40	1.60	100	2.0	29	2.60	40	3.80
300	9/16	169.0	91	1.33	1.50	1.80	2.20	150	3.0	43	3.85	60	5.90
400	9/16	225.0	1.21	1.76	1.95	2.40	2.80	200	4.0	57	5.35	80	7.40
500	9/16	282.0	1.52	2.20	2.50	3.00	3.40	250	5.0	71	6.60	100	9.25
600	9/16	338.0	1.83	2.64	3.00	3.50	4.00	300	6.0	86	7.60	120	11.10
700	9/16	394.0	2.13	3.08	3.50	4.00	4.60	350	6.9	100	8.80	140	12.40
800	9/16	450.0	2.43	3.50	3.90	4.50	5.20	400	7.8	114	10.00	160	14.20
900	9/16	507.0	2.73	3.92	4.30	5.10	5.80	450	8.6	129	10.70	180	15.90
1000	9/16	563.0	3.03	4.34	4.70	5.70	6.50	500	9.4	143	11.90	200	18.90

The Lower Pressures are used with Shorter or Smaller Cans.

pressure being used on the smaller cans and 5 lbs. being used on the larger cans. There is some difference between the amounts of air required by the high- and low-pressure systems, the difference being in general that the high-pressure system may require the smaller amount of air.

Some additional data on volume of air and horsepower required for agitation is shown in Table 77.

Distilling System.—Originally, ice was manufactured in commercial quantities by means of the distilled water system. In this type of plant, the apparatus consists principally of a steam boiler, a steam engine, the compressor, the distilling system and the freezing system. The water, after being taken from a suitable source of supply, is put into a steam boiler in which it is evaporated into steam for the purpose of driving the main steam engine. The steam engine is generally of the Corliss type and considerations of steam economy therefore are not important, since it is generally desirable to allow the steam engine to produce enough exhaust steam which, after being condensed, will supply sufficient water for making the ice.

The exhaust from the main steam engine, as well as from the auxiliaries, is led through an oil separator. The function of the oil separator is to remove all particles of oil which are carried along with the steam from the engine. It is important to remove as nearly as possible all oil or grease from the steam before it enters the steam condenser. After the steam has been freed from all traces of oil and grease, it then enters the steam condenser. In the ordinary ice making plant the function of the steam condenser is to simply convert the exhaust steam into water. It is, therefore, undesirable to try to maintain a vacuum in the condenser. Steam condensers for performing this function may be constructed either in the atmospheric type of the galvanized sheet iron type.

The condensed water is now led to the reboiler and skimmer. The function of this apparatus is to boil the water at atmospheric pressure to drive off any air or gases which may be contained in the water. Oil, grease, or other impurities rise to the top of the water and are skimmed off, then allowed to drain to the sewer. The reboiler and skimmer are generally constructed in the cylindrical tank type and the rectangular type. Live steam from the boiler is admitted into suitable pipe coils which are submerged in the water. This causes the water to be reboiled, and thereby purified. The height of the water in the reboiler tank may be regulated by means of a float which is connected to a lever handled gate valve in the distilled water line at a point below the water in the storage tank.

The hot water from the reboiler and skimmer is next led to a suit-

able water cooler. The function of this apparatus is simply to lower the temperature of the water down to within a few degrees of the temperature of the available cooling water. The apparatus may be constructed in either the atmospheric or the double-pipe type. In the atmospheric type the hot water from the reboiler is circulated through a vertical pipe coil, while the cold water is allowed to flow over the outside of the pipe. The cooled water is next led to suitable filters. These filters usually contain charcoal, quartz or sand, which completes the process of purification of the water by removing any impurities which would cause the ice to have an odor or taste.

After the water has been filtered and purified in this manner it is next led to a storage tank. The level of the water in the storage tank is controlled by means of a float valve. The storage tank should be made large enough to contain a supply of water for a convenient length of time. The water is conducted from the storage tank through a rubber hose and can filler into the ice cans. In some distilled water ice making plants it has been the practice to place a coil of pipe in the water storage tank and insulate same for the purpose of cooling the water to a lower temperature. The pipe coil in the water may be operated either on the direct-expansion system or by allowing the return vapors from the ice-making tank to pass through it. It is generally considered advisable to use the direct-expansion system, instead of the return vapor system. This is due to the fact that unless the expansion valves on the evaporating coils are regulated properly the return vapors may become superheated upon passing through the water storage tank.

Probably the best method of cooling the water is to shower the water over the outside of the vertical pipe coils which are fitted for the direct expansion of ammonia. By this means the water may be completely precooled to 33° or 34° F. The advantage of this system is obvious and is due to the fact that this amount of refrigeration work is performed on the outside of the ice-making tank. This will allow the ice-making tank to produce more ice.

In the larger distilled water ice-making plants it is sometimes necessary, on account of economical considerations, to use compound condensing steam engines instead of single engines. This arrangement is used in order to produce more ice per ton of fuel. In the larger plants it is, therefore, necessary to use compound condensing engines which have a low fuel consumption and a proportionate low steam consumption. In this case, the exhaust steam from the steam engines is not sufficient for ice making purposes and an additional amount must be supplied by a suitable evaporator.

In this type of plant, the additional apparatus consists of an evaporator, a steam condenser with an air pump, and a vacuum reboiler.

The exhaust steam, after passing through the oil and grease separator, enters the evaporator under a vacuum. The evaporator is, in turn, connected with a regular steam condenser which operates under a much higher vacuum. The maintaining of a high vacuum on the condenser side of the evaporator makes it possible to evaporate the water in this portion of the evaporator at a lower temperature. The whole equipment is so regulated that the temperature in the evaporator on the high vacuum side is such that the incoming steam from the oil and grease separator is condensed, and at the same time the temperature in the evaporator portion which is under low vacuum will produce the necessary additional water vapor.

The original exhaust steam from the oil and grease separator, together with the additional vapor formed in the evaporator, passes on to the steam condenser, where it is converted into water. The water, after being reboiled in a vacuum reboiler, flows either by gravity or through an automatic pump to a skimmer. The rest of the cycle of operation of the plant using the evaporator is identical with the one previously described.

The arrangement of the distilling apparatus on the small or medium-sized plant is shown by Fig. 148. In this system, the water drains by gravity from the steam condenser through the various parts of the apparatus to the storage tank. In some cases, it is not possible to locate the steam condenser at such an elevation that would produce the gravity flow of water. In this case, it is necessary to pump the water from the steam condenser to the reboiler, from which it drains by gravity through the coolers and filters to the storage tank. In Fig. 148, it will be noted that the steam, after passing through the oil and grease separator, passes directly into a galvanized sheet-iron steam condenser. This condenser is usually termed the flask condenser. The cooling water is showered over the outside of the condenser, thereby condensing the steam on the inside. The condenser is protected from excessive pressure by a suitable relief valve. They are also generally provided with suitable vents for purging off non-condensable gas.

The condensed steam flows to the reboiler, which is of the cylindrical tank type. The reboiler float tank is connected directly to the reboiler so that the water level in the reboiler may be regulated to a certain height. The reboiled water passes next to an atmospheric distilled water cooler and after being cooled to a temperature varying from 70° to 90° F., it passes through the charcoal filters to the storage tank.

Storing of Ice.—After the ice has been manufactured, as previously indicated, it is stored in suitable rooms until it is ready to be sent to the

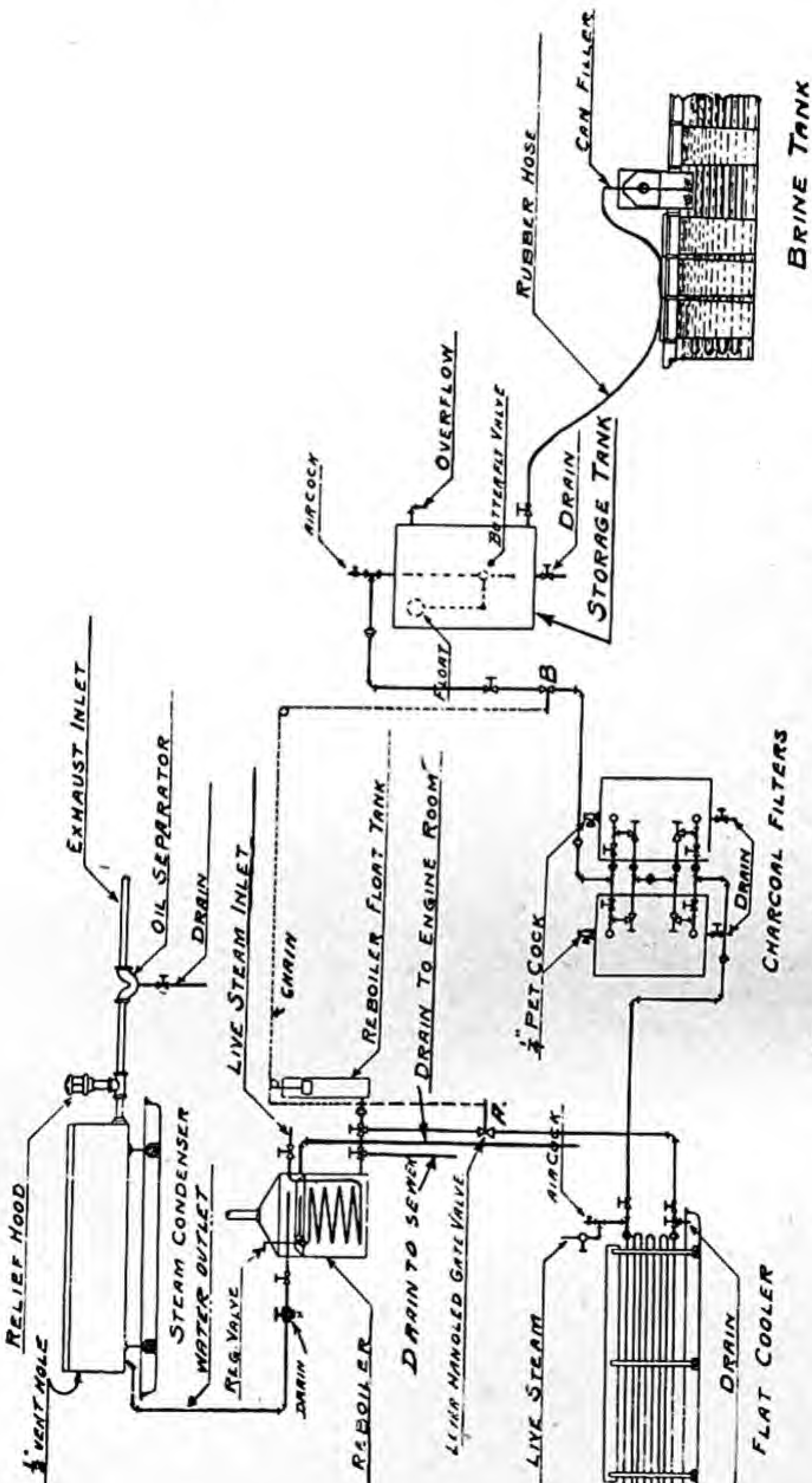


Fig. 148.—Distilling System, Gravity Type.

ultimate consumer. The ordinary ice plant should have a small daily ice storage room located near the ice tank into which the ice is dumped from the cans. The size of this room should be such that it will hold from three to ten times the output of the plant per day. The floor area for the daily ice storage room, in which the blocks of ice are generally placed on end instead of being stacked, may be estimated by allowing 14 sq. ft. per ton of ice for 300-lb. blocks, and 11 sq. ft. for 400-lb. blocks. These figures will allow sufficient room for aisles, etc. Since the ice blocks are stood on end only, the height of the ceiling need not be great and in general should range from 8 to 10 ft.

Daily ice storage rooms may be refrigerated by means of direct-expansion piping, or brine piping. In the more modern plants, direct-expansion piping is used for refrigerating the ice storage. However, under certain local conditions, depending upon the layout of the plant, etc., brine taken from the ice freezing tank may be circulated through the coils in the ice storage room for maintaining the proper temperatures.

The proposition of providing a yearly storage into which the output during the slack season is stored, is one of economic importance. In general, the proposition must be analyzed from all possible viewpoints, giving especial attention to the desirability of using an ice storage, the design of the building for ice storage, the economical means for refrigerating same, etc.

Shut-downs of short periods due to mechanical difficulties of operation may not prove so disadvantageous when the plant is equipped with the proper storage capacity.

In order to determine the most economical size of a storage room for a given plant, such factors as the demand for ice throughout the year, the greatest demand during the summer, the probabilities of future expansion, etc., must be taken into consideration. The principal determining factor in the consideration of the size of an ice storage is the distribution of the ice sales throughout the year. Table 78 has been prepared to show the relation of the ice sales distribution and the magnitude of the ice storage room for a plant which is manufacturing 25 tons of ice per day.

Column 3 gives the distribution of ice sales throughout the various months of the year. Column 4 indicates the full load capacity of 25 tons per day. Column 5 shows the estimated sales per month. Column 6 indicates the shortage that would occur during the months of maximum demand for ice. Column 7 indicates the proper amount of storage in the months of smaller demands in order to offset the shortage.

After the amount of ice storage has been determined in a manner similar to the foregoing, the number of cubic feet of capacity in the ice

TABLE 78.—ICE SALES, DISTRIBUTION AND STORAGE.

Month	Days	Distribution	Capacity	Sales	Shortage	Storage
January	31	2%	775	183	...	592
February	28	2%	700	183	...	517
March	31	3%	775	274	...	501
April	30	5%	750	457	...	293
May	31	8%	775	730	...	45
June	30	15%	750	1,368	618	...
July	31	17%	775	1,551	776	...
August	31	17%	775	1,551	776	...
September	30	14%	750	1,277	527	...
October	31	8%	775	730	...	45
November	30	6%	750	547	...	203
December	31	3%	775	274	...	501
Total	365	100%	9,125	9,125	2,697	2,697

storage room may be estimated by allowing 45 to 50 cu. ft. per ton of ice stored. The preferred dimensions of the ice storage room should be as nearly as possible to those of a cube. These dimensions give a room with the largest cubic capacity and with the least amount of wall, floor and ceiling area.

Any of the up-to-date materials of construction may be used for constructing the ice storage room. Brick, concrete and wood are commonly used. Ice storage rooms made with brick have the walls strengthened by building pilasters on the outside. The thickness of the wall and the detail of design of same depend upon the height of the wall, the layout of the plant, etc. The foundation and floor, of course, may be constructed of concrete. The roof is supported by steel girders which in turn are supported by the pilasters and the walls. The ceiling proper may be made of concrete and supported by means of beams between the lower members of the girders. Reinforced concrete construction may be used throughout the room, in contradistinction to brick and wood. The ice storage room may also be constructed by the use of concrete blocks or hollow tile in combination with the steel skeleton to support the roof. The ice storage room may be constructed of timber in a similar manner.

The ice storage room should be insulated on the ceiling, floor and walls with 4-in. cork or mineral wool boards. The insulation should be attached to the ceiling construction in an efficient manner, and all insulation should have the joints broken so as to reduce the loss of refrigeration to a minimum. The wall and ceiling surfaces should be finished with emulsified asphalt. The floor may be constructed by putting down a concrete base, after which the sheet insulation is applied with hot asphalt. A wearing surface of a few inches of concrete is then added over the insulation.

The temperatures of the ice storage room should be maintained at 26° to 28° F. The refrigerating piping should be laid out in a manner so as to distribute the refrigeration about the room in the proper manner.

The handling of ice in the storage room should also receive attention. The daily ice storage room, the yearly ice storage room, and the loading platform should be laid out with the idea of reducing the labor required for the handling of the ice to a minimum. The daily storage room may be equipped with mechanical conveyors for moving the ice from the room to the loading platform. The yearly storage room should be equipped with the proper number of ice elevators, or in the event that the room is not high enough to warrant the installation of the elevators, ice piling machines should be used.

QUESTIONS ON CHAPTER XII.

1. Describe and name the advantages of the low-pressure air system as used in raw water ice making plants.

2. Describe and name the advantages of the high-pressure air system in raw water ice plants.

3. Explain fully how impurities in water affect the water during the freezing process.

4. Name some of the common impurities in water which is used for ice making, and describe the effects of such impurities.

5. Describe the methods employed for the removal of common impurities in the water.

6. Name and describe the various factors which affect the amount of air required for agitating the water during freezing process.

7. Describe fully the distilled water ice making system.

8. Since ice freezes faster as the temperature is lowered, why is it not advisable to increase plant output by freezing ice at $-20°$ or $-30°$ F.?

9. Determine the size of a yearly ice storage room for a 100-ton ice plant having an ice sales distribution as shown by Table 78.

10. Why does the white core of an ice block have a flavor different from that of the water from which the ice is frozen?

CHAPTER XIII.

COOLING SOLIDS, LIQUIDS AND GASES.

Creamery and Dairy Refrigeration.—The production of milk, butter, cheese, and ice cream is increasing each year. The production of milk runs into hundreds of billions of pounds, sufficient to provide each person in the country with two and one-half to three pints of milk daily.

Production of butter in the United States has shown a more marked upward trend than population. There has been a pronounced shift from the production of butter on farms to factory production. This change has been due to the necessity of increasing the supply and of more sanitary methods of production. Refrigeration has been a prominent factor in both instances.

Improvements in the manufacture of cheese have very materially increased the manufacture of this essential food in the United States; the annual production amounting to practically a half billion pounds. The number of factories engaged in producing cheese show a steady increase from year to year.

The manufacture of ice cream has shown a great increase during the past five years and this is due largely to more efficient methods of production in which mechanical refrigeration has played the most prominent part; the number of ice cream plants throughout the United States shows a steady increase from year to year. In 1930 the estimated production of ice cream in the United States amounted to 345 million gallons. This would be equivalent to 2.8 gallons per person a year.

Mechanical refrigeration is a necessity in keeping milk at a proper temperature and also in the process and manufacture of other commodities in which milk is the base. United States Government inspection, sanitary, and other regulations in the various states has made it obligatory that milk and milk products shall be maintained at temperatures which will preserve their nourishing qualities and keep them safe for human consumption.

Need of Refrigeration.—From the foregoing, it will be observed that the creamery and dairy industry is one of the most important of the country, since it supplies daily foods. The need of refrigeration in the transportation and distribution of the milk supply is apparent. The sources of the supply are quite distant in general from the more thickly populated districts such as the large cities, so that the time required to transport the milk will vary from 36 to 48 hours. In this case, if the temperature of the milk is allowed to rise to 60° F., or above it will become badly contaminated. To reduce the danger of disease, the milk is pasteurized, after which it is cooled to a temperature of 35° to 40° F., to prevent the growth of bacteria. After being bottled or put into suitable cans, it may be stored a short time at this temperature, after which it is distributed to the ultimate consumer.

Of the total amount of butter produced in the United States, about 95 per cent is consumed within a short period after production. This period will vary from a few days to three months. The remaining 5 per cent is held in cold storage warehouses at a low temperature for periods varying from three to fifteen months. The storage of this amount of butter helps to keep the price of the butter more uniform during the winter months, when there is a scarcity of butter.

The need of refrigeration for producing ice cream is apparent. In fact, ice making and ice cream making require very near the same kind of refrigeration work. This, in general, amounts to cooling of the milk or water to the freezing temperature, the freezing of the material, and the lowering of the material to a low temperature, together with other losses which occur in the plant. Refrigeration is used in cheese factories not only for the purpose of cooling milk and water, but also for the cold curing of cheese.

From the foregoing, it will be appreciated that the use of mechanical refrigeration in the manufacture of these various commodities is a necessity, not only from the sanitation viewpoint, but also from that of efficient means of production.

Typical Creamery and Dairy Plant.—Many commercial plants are devoted exclusively to the production of one particular commodity, either milk, ice cream, or cheese. However, in many commercial plants, since the manufacture of the various products is so closely related, two or more are produced within the same plant. Such a plant generally produces cooled and pasteurized milk, cream, butter and ice cream for the ultimate consumer. One of the small typical creamery and dairy plants used for the production of milk, ice cream, butter, etc., is shown by Figs. 149, 150 and 151. Fig. 149 represents the basement plan of such a factory. The mechanical equipment, such as steam boilers, steam engines, electric motors, compressors, and

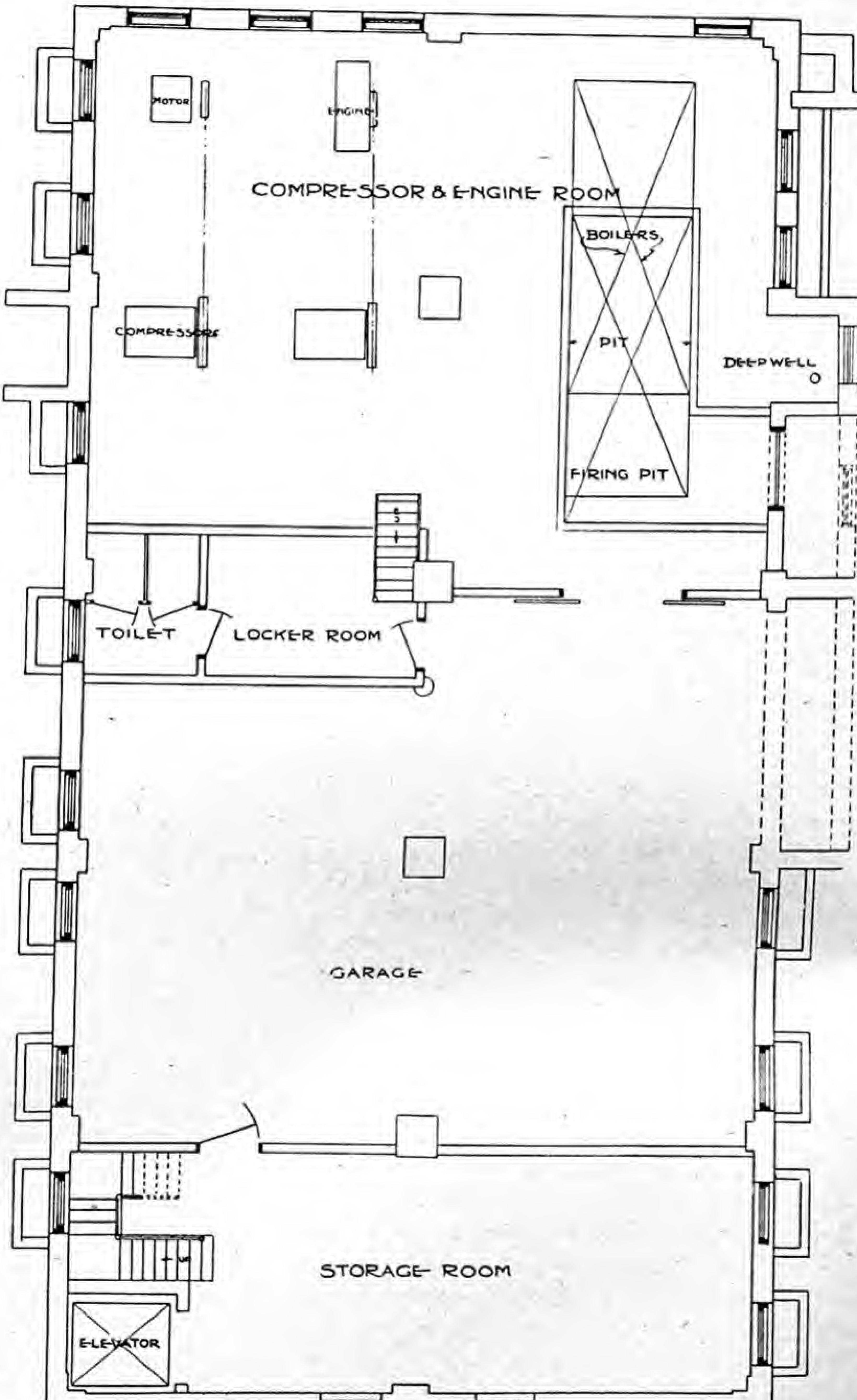


Fig. 149.—Basement Plan of Typical Creamery and Dairy Plant.

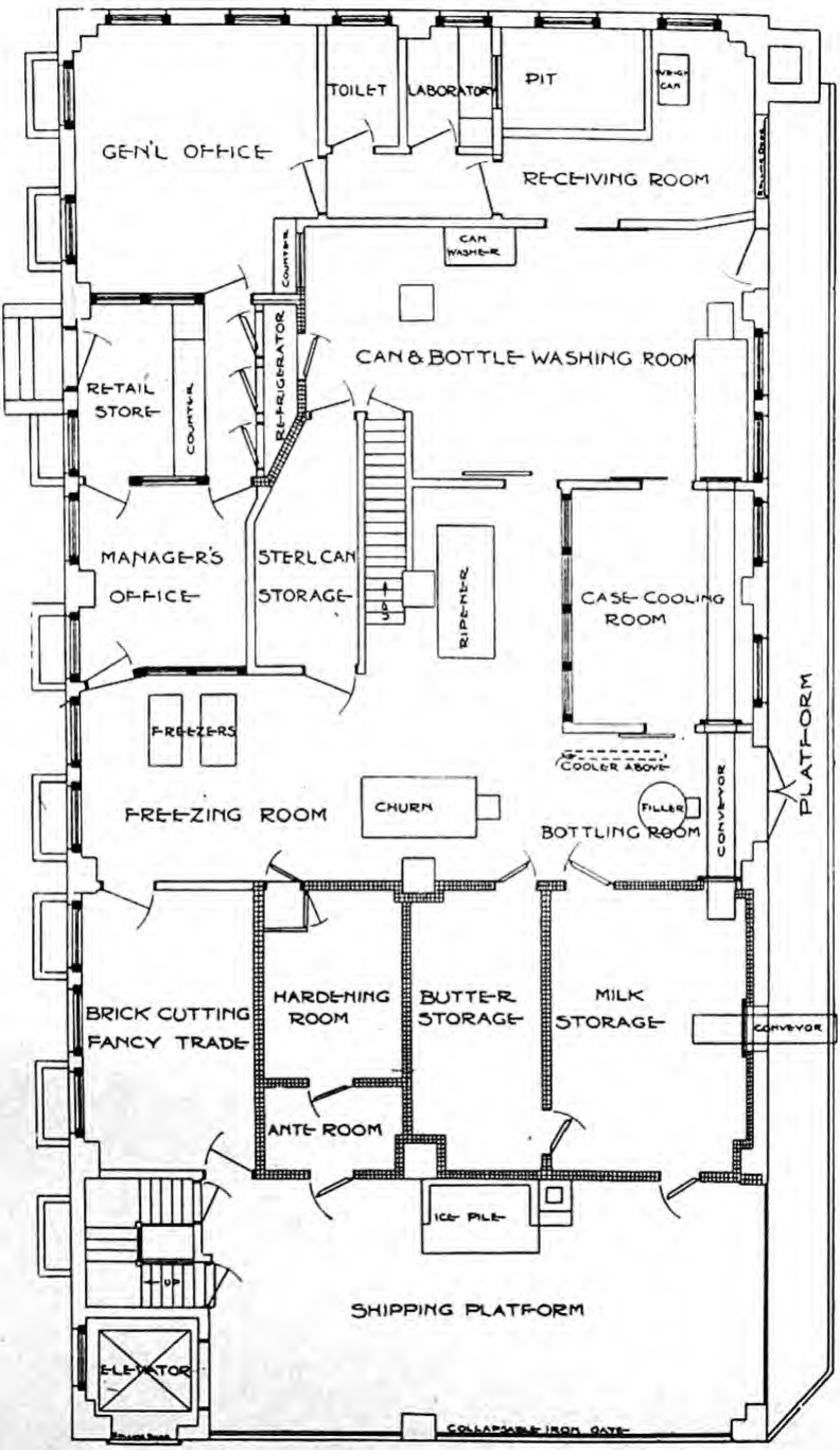


Fig. 150.—First Floor Plan of Typical Creamery and Dairy Plant.

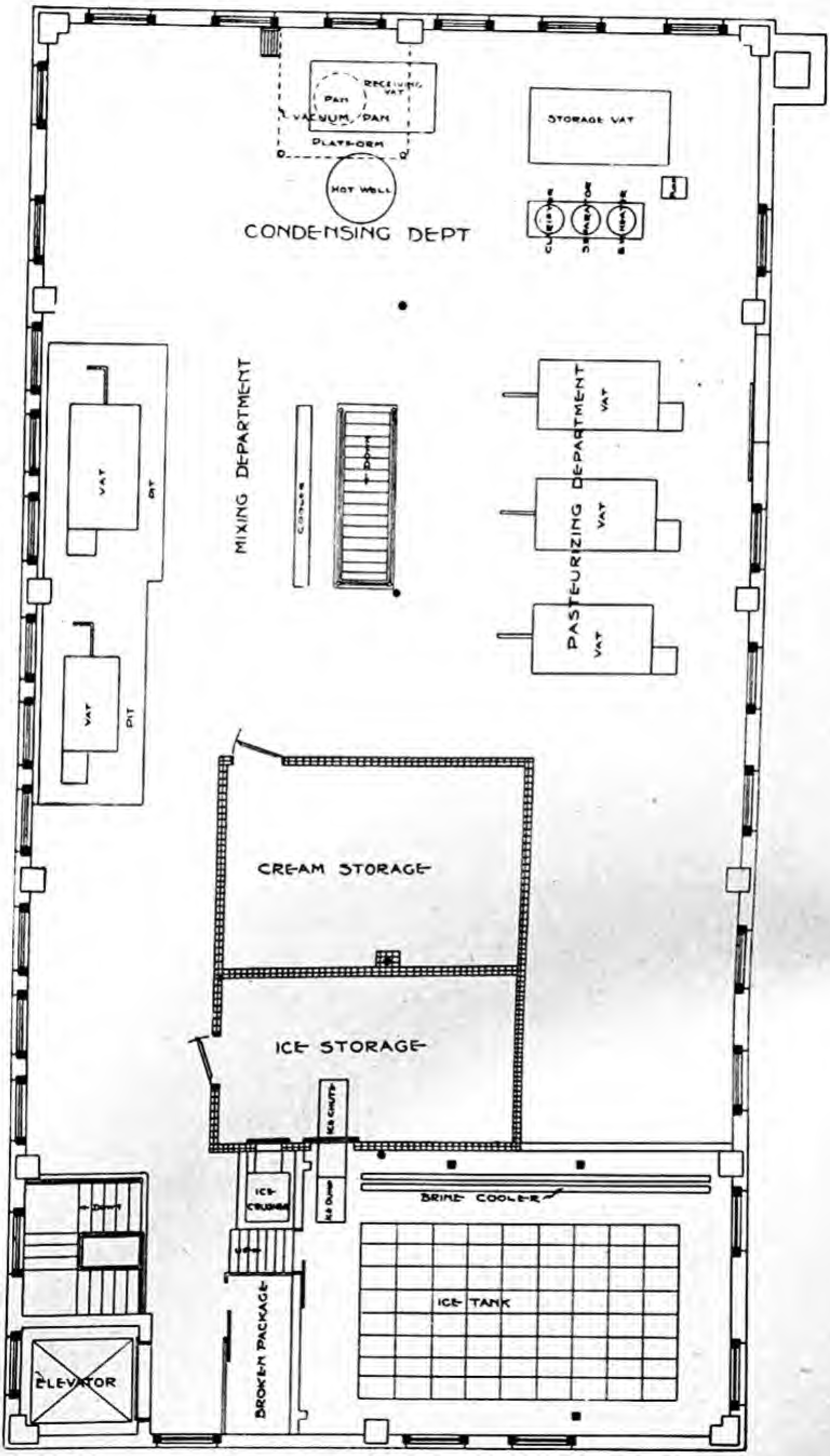


Fig. 151.—Second Floor Plan of Typical Creamery and Dairy Plant.

other refrigerating equipment, are located in the basement. The first floor contains the ice cream freezing room, the ice cream hardening room, the butter storage and the milk storage, together with a small refrigerator for retail trade. The second floor contains a cream storage, ice making tank, and ice storage room. The various auxiliary apparatus used in the condensing department, the pasteurizing department, the ice cream mixing department, are located at suitable points.

From the foregoing, it will be observed that there are many points about the factory where refrigeration must be used. The refrigeration per day required in a small typical plant of this size is listed in the following tabulation:

(1) To cool 1500 gallons of milk from 80° to 40° F.
(2) To cool 3750 pounds of ice cream mix from 80° to 40° F.
(3) To cool 200 gallons of milk from 80° to 60° F.
(4) To cool 500 gallons of cream from 80° to 50° F.
(5) To cool 500 gallons of buttermilk from 80° to 60° F.
(6) To freeze and harden 750 gallons of ice cream.
(7) To make six tons of ice.
(8) To cool 400 gallons of water from 80° to 40° F.
(9) To maintain the hardening room at 0° F.
(10) To hold the ante-room to the hardening room at 32° F.
(11) To hold the butter storage room at 32° F.
(12) To hold the milk storage room at 35° F.
(13) To hold the cream storage room at 30° F.
(14) To maintain the ice storage room at 28° F.
(15) To cool the small retail sales refrigerator.

Refrigeration for Cooling Milk, Cream and Buttermilk.—In order to eliminate as nearly as possible the bacteria contained in milk, the milk is pasteurized. This consists of either bringing the milk up to a temperature of 140° or 150° F. and holding it at this temperature for 20 or 30 minutes, or raising the temperature to 160° or 165° F., and holding it at this temperature from one-half to one minute. The first process is preferable and is the one that is generally used. After the milk has been pasteurized it should be reduced in temperature to 35° or 45° F. as soon as possible. This cooling should occupy from one to two hours. The cooling of the milk from the high temperature to a lower temperature of 80° to 85° F. is accomplished by means of the use of cool water. Refrigeration is generally used to reduce the temperature below this. Sometimes, cold brine is used to cool the warm milk coming from the pasteurizer. This imposes an additional and unnecessary load upon the refrigeration equipment.

Figure 152 shows the arrangement of a milk cooler which uses water and brine for cooling the milk. In this cooler, the milk at a high temperature coming from the pasteurizer is allowed to flow over the outside of the vertical pipe coils. The upper pipes are cooled by water which lowers the temperature of the milk to 80° to 85° F. The

lower section of the pipe coil may be cooled by cold brine or cold water which reduces the temperature of the milk to 35° or 40° F. Another type of milk cooler is shown by Fig. 152a. In this apparatus, the cold raw milk is heated in a heat exchanger by the hot milk coming from the pasteurizer. The milk is further cooled by cold brine or water in the lower section of the vertical pipe coils. In addition to these types of coolers, double pipe horizontal helical coil coolers may be used. Direct expansion milk coolers may be used also. As previously indicated water is used to cool the milk (or cream) to 80° to 85° F.

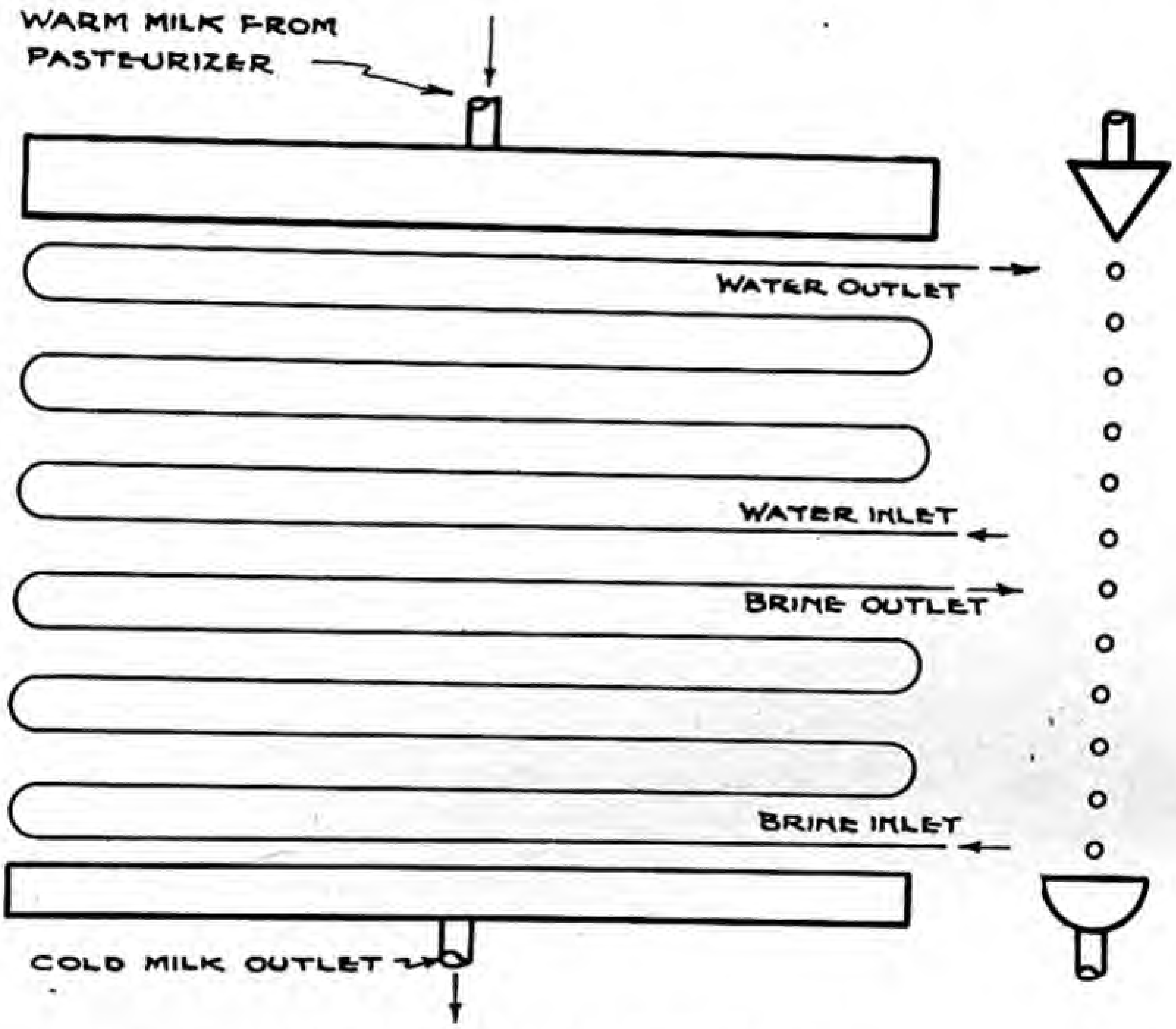


Fig. 152.—Overflow Milk Cooler.

From the milk coolers, the milk generally goes to a bottle filler or is filled directly into milk cans. It is then put into the storage room, or is shipped from the factory. By the time it has been bottled or put into the cans, the temperature will have risen from 35° or 40° F. to 60° or 70° F., so that in estimating the total amount of refrigeration, a double cooling effect must be allowed if the milk is put into the storage room immediately after being bottled or put into the cans. Due to the fact that the cooling of the milk after it has been pasteur-

ized must be rapid, the refrigeration requirement during this period is quite large. It is, therefore, generally necessary to provide a large quantity of brine for the purpose of absorbing this heat. Under this condition, the refrigerating machine would have a smaller capacity and would operate throughout the entire day if necessary.

The refrigeration required to cool a given amount of milk or cream in a given time may be estimated by using the following formula:

$$H = W \times S \times (t_1 - t_2)$$

where H = heat to be removed
W = weight of cream or milk in lbs.
S = specific heat Btu. per lb.
t_1 = higher temperature
t_2 = lower temperature

The weight of milk is approximately 8.6 lbs. per gal., while the weight of cream is about 8.4 lbs. per gal. The specific heats of milk and cream may be taken from Table 83, Chapter XV. The refrigeration required to cool 1,000 gals. of milk from 80° to 40° F. would be calculated as follows:

$$H = 1000 \times 8.6 \times 0.90 \times (80 - 40) = 309{,}600 \text{ Btu.}$$

If this cooling is produced by allowing brine to rise from 8° to 28° F., it is evident that a certain quantity of brine would be required. This would depend upon the specific heat of the brine and the weight per cu. ft. In the case that the specific gravity of the brine is 1.20 and the specific heat is equal to 0.70, the cu. ft. of brine required may be calculated as follows:

$$\begin{aligned} 309{,}600 &= W \times 0.70 \times (28° - 8°) \\ W &= 309{,}600 \div (0.70 \times 20) \\ &= 22150 \text{ lbs.} \\ \text{cu. ft.} &= 22150 \div (62.5 \times 1.2) \\ &= 295 \end{aligned}$$

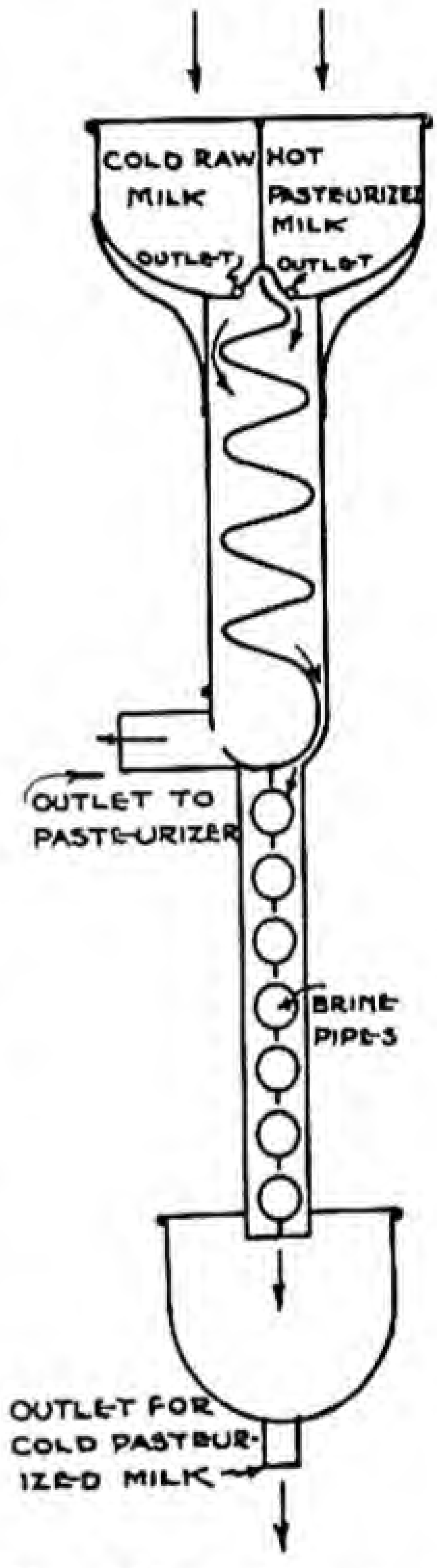


Fig. 152a.—Counter-Flow Milk Cooler.

The refrigeration required for cooling the cream is calculated in a similar manner. The refrigeration required to cool buttermilk may be estimated to cool the equivalent weight of water, since the buttermilk contains a very large percentage of water.

Refrigeration for Ice Cream Making.—As previously indicated, the refrigeration required for making ice cream is very similar to that for freezing ice. Details of the operation are somewhat different, and much lower temperatures are required in the production of ice cream.

The ice cream mix is composed of various constituents, depending upon the quality of cream desired, the flavor, etc. The ice cream mix is generally made up and then stored for from 24 to 48 hours. The following tabulation gives an idea of the composition of ice cream mix:

Ingredients	*Per cent solids*
11.0 lbs. sugar	10.45
1.0 lbs. filler and flavor	0.50
16.66 lbs. butter 84%	14.00
8.45 lbs. skim milk powder	8.05
62.89 lbs. water	00.00
100.00 lbs. mix	33.00 total solids

Ingredients	*Per cent solids*
11.0 lbs. sugar	10.45
1.0 lbs. filler and flavor	0.50
13.2 lbs. butter 84%	11.08
72.8 lbs. milk 4%	9.13
2.0 lbs. skim milk powder	1.90
100.00 lbs. mix	33.06 total solids

The ice cream mix, after having been stored for a short time, is put into the freezers. The ice cream freezers may be cooled by either direct evaporation of liquid refrigerant or by cold brine. The freezers consist generally of a horizontal cylinder around which is a jacket for the refrigerant. The ice cream mix is put into the cylinder at a temperature of 35° to 50° F. Suitable dashers whip the cream during the freezing process. The ice cream freezers are made in various sizes, varying from 10 to 25 gals. per freezing. The ice cream mix is allowed to remain in the freezers for 5 to 10 minutes, at the end of which time it is in a state similar to thick syrup. The temperature required in the cooling jacket of the ice cream freezers varies with the nature of the cream being frozen and in general will vary from −15° to 5° F. During the process of freezing the ice cream mix swells about 70 to 80 per cent in volume due to the whipping and freezing. The condition of the mix, the relative temperature, size of batch, speed of dasher, etc., affect the relative amount of swell or overrun.

Direct-expansion ice cream freezers may be used also. These eliminate brine coolers, pumps, and lines. A representative type of direct-expansion freezer is shown by Fig. 153.

The extent of the expansion of the cream during the freezing process depends upon the foregoing factors. After the cream has been partly frozen in the freezer it is withdrawn and put into cans. These cans with the ice cream are then stored in a room of low temperature. The temperature in these rooms will, in general, be about 0° F. However, in some instances hardening room temperatures as low as −30° F. have been used. The storing of the cream in the room

of low temperature allows the cream to freeze entirely into a solid state. In small plants, where the capacity is quite low, the ice cream may be hardened by submerging the cans into a tank of cold brine. In plants having larger capacities the still-air hardening room may be used.

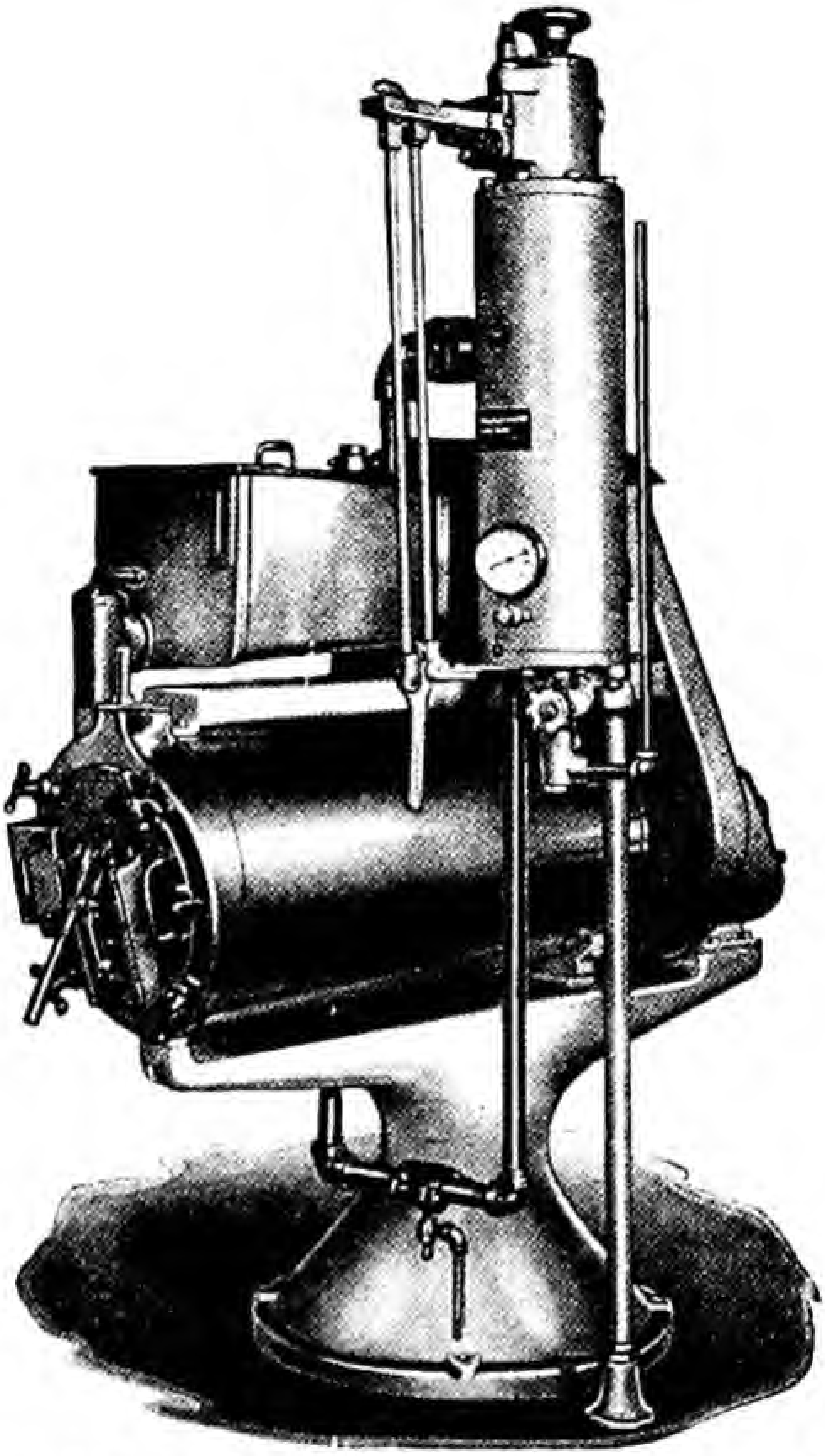


Fig. 153.—Creamery Package Ammonia Ice Cream Freezer.

The hardening rooms are generally refrigerated by means of direct-expansion pipes, which are arranged in the form of shelves. The cream to be hardened is placed directly on these shelves and allowed to remain in the hardening room from 36 to 72 hours. In the hardening rooms which employ forced air circulation, the direct-expansion pipes are generally located in a bunker. A fan is used to circulate the air across the coils and then down around the ice cream cans. Still-air hardening rooms will, in general, require about 1½ to 2 cu. ft. of cubic

capacity per gallon of ice cream produced per day. This will allow 2 to 3 days' storage of the daily production.

From the foregoing it will be noted that the refrigeration consists of cooling the raw material, the removal of part of the latent heat of fusion and some cooling in the freezer, the removal of the rest of the latent heat of fusion and the final cooling in the hardening room, the removal of heat transmitted by insulation and other losses.

From the nature of the process it will be observed that it is more difficult to estimate the exact refrigeration requirements than when ice is made. This is due to the fact that the specific heat of ice cream before freezing, the latent heat of fusion, and the specific heat of the ice cream after being frozen will vary with the kind of cream to be produced. Approximate figures for the specific heat and the latent heat of fusion of ice cream may be taken from Table 83 of Chapter XV.

The weight of ice cream after it has been frozen will depend upon the swell or the increase of volume during freezing and the specific gravity of the mixture. In the event that the swell is 75 per cent and the specific gravity is equal to 1.10, the weight of a gallon of ice cream would be calculated as follows:

$$\left(\frac{1}{1.00 + 0.75}\right) \times 8.33 \times 1.10 = 5.24 \text{ lbs.}$$

In general, it may be said that the ice cream containing only extract flavoring will weigh approximately 5 lbs. per gal. after it has been frozen; ice cream containing fruit, nuts, etc., will weigh somewhat more, generally about 6 lbs. per gal. Using the values for the specific heats and the latent heats of fusion given in Table 83 of Chapter XV, the refrigeration required to freeze and harden a gallon of ice cream from a temperature of 50° F. to a temperature of 0° F. may be calculated as follows:

Cooling to freezing point........	$1 \times 0.78 \times (50 - 26) =$	18.72 Btu.
Latent heat of fusion............	$1 \times 90 \qquad =$	90.00
Cooling after freezing...........	$1 \times 0.45 \times (26 - 0) =$	11.70 Btu.
Total refrigeration per lb.		120.42

If the ice cream weighs 6 lbs. per gal. it would, therefore, require $6 \times 120.42 = 722.5$ Btu. per gal. The loss of refrigeration through the walls of the cold rooms could be estimated by the methods outlined in Chapter VIII. After this and other losses have been determined, the refrigeration required must be based upon the foregoing calculation—that is, 722.5 Btu. per gal. For small plants, additional refrigeration required to offset the heat transmission through the insulation and other losses may be estimated as being equal to that required for cooling and freezing the cream. In this case, the total refrigeration

could be estimated by allowing $722.5 + 722.5 = 1445$ total Btu. per gal. This is in line with the old rule of thumb which stated that a one-ton refrigerating plant running 12 hours per day would produce refrigeration for 100 gals. of ice cream.

$$\text{This would be equivalent to } \frac{288{,}000}{2 \times 100} = 1440 \text{ Btu. per gal.}$$

The refrigeration required for the ice cream freezer may be taken as the heat required to cool the ice cream mix to the freezing point and one-half the latent heat of fusion. The refrigeration in the ice cream hardening room may be taken as being the other half of the latent heat of fusion, together with the amount of heat required to cool the ice cream from the freezing point to the temperature of the hardening room. After the heat to be absorbed in the ice cream freezer has been determined, the quantity of brine to be circulated may be estimated; and in a similar manner, after the heat to be removed in the hardening room has been determined, the amount of direct-expansion piping to be installed in the ice cream hardening room may be estimated.

Of course, in addition to cooling the ice cream, the heat that is transmitted through the insulation in the ice cream hardening room, as well as the heat entering with the warm air from the outside, must be removed.

Under general plant conditions, when ammonia is used as the refrigerant, a back pressure of approximately 5 lbs. must be carried in order to maintain the hardening room at a temperature of 0° F.

Ice that is required for packing purposes will vary with the local conditions at the plant and the size of the plant, and in general will range from 10 to 20 lbs. per gal. of ice cream manufactured. Under general plant conditions, it will be advisable to manufacture the ice that is needed in the ice cream manufacturing plant itself when the capacity of the plant is 750 or more gals. per day. Under this same condition, the compressor working on the ice tank should be operated throughout the day. From the foregoing it will be noted that comparatively low temperatures are required in the ice making factory, as well as comparatively high temperatures. This often makes it necessary to install a refrigerating system which will operate at two back pressures.

The refrigeration for ice cream freezing and hardening rooms may be produced by the low pressure machine, while that required for cooling the cream, milk, water, and for ice making may be produced at the higher suction pressure. The brine for circulation through the ice cream freezers may be cooled either in the ice making tank or in a special cooling tank. If the brine is cooled in the ice making tank it

is necessary to keep the brine temperature quite low, which makes it necessary at times to install a special tank for cooling the brine for ice cream freezers. Refrigeration is generally produced in this sort of tank by means of submerged direct-expansion coils.

It is also advantageous at times to install a double-pipe brine cooler for the purpose of assisting the brine cooling tank or the ice making tank in cooling the brine. The brine pump is used to circulate the brine through the brine cooling tank and the brine cooling apparatus to the freezers and other parts of the plant that must have brine refrigeration.

Water Cooling Systems.—Mechanical refrigeration is being used extensively for the cooling of water in industrial plants. Cooled water is necessary in the manufacture of soft drinks, drinking water systems, and various other commercial uses. Drinking water systems for supplying cooled water for drinking purposes are now installed in hotels, hospitals, theaters, schools, factories, office buildings, department stores, etc. Cooled water is used for commercial uses in such industrial plants as paper manufacturers, rubber manufacturers, bakeries, chemical factories and soft drink manufacturers.

Drinking Water Systems.—Due to the demand for more sanitary conditions about factories, mills, foundries, as well as in hotels, office buildings, the older type of water distribution systems has been superseded by the drinking water system which supplies water cooled by mechanical refrigeration. After the water has been taken from the source of supply, it is passed through a cooler and then into the water distribution lines. These are generally covered with some efficient insulator, and at suitable intervals fountains are attached to these water lines. The advantages of this method of water distribution for drinking purposes are obvious. It supplies water that contains a minimum of bacteria and which is attractive in appearance. It eliminates the spread of disease which might be transmitted through the use of common drinking vessels; it supplies water at a proper temperature for the body; it reduces the time required for workmen, employes, etc., to get water.

Water for drinking purposes may be cooled in general by two systems. The first system is known as the open system and is shown diagrammatically by Fig. 154. This is known as the open system because the water is cooled by allowing it to flow in an open flow over a vertical pipe coil which contains a suitable refrigerant at low temperature. The water is admitted to the plant from the city supply mains through a filter. The amount to be admitted is controlled by means of a float valve connected to a float in the water cooling tank. This maintains the water at a predetermined level in the tank. After the water has

been cooled by flowing over the cold refrigerating coils it is led to a circulating pump, which discharges the cooled water into the distributing system. This generally consists of a loop of piping, arranged

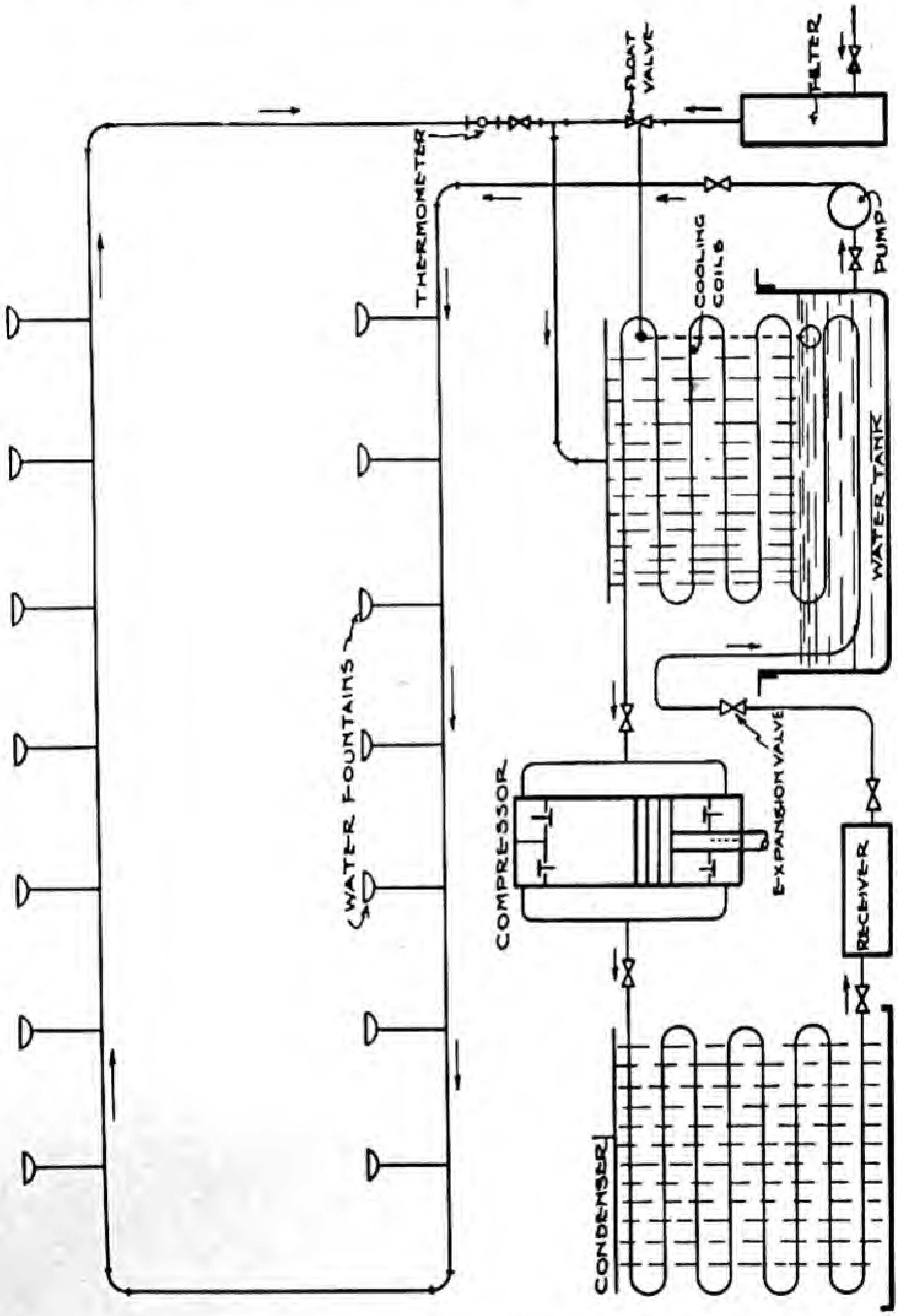


Fig. 154.—Open Type Water Cooling System.

about the building. These loops of piping generally are made of pipe varying in size from one inch to two inches, depending upon the length, the quantity of water used.

At suitable intervals upon this loop of piping the fountains are attached at which the workmen may get cooled, clear water. Any

waste water is allowed to drain to the sewer. After the water has passed through the loop it is again returned to the water cooling tank, and, with the fresh water from the city mains, it is cooled by allowing it to flow over the refrigerating coils again. A pressure of 10 to 15 lbs. is maintained at the highest point in the system by applying a relief valve to the water line just before it returns to the water cooling tank. The refrigeration may be produced by the evaporation of any suitable refrigerant. In the case of the use of ammonia as the refrigerant, the coils are generally made of continuously welded pipes to reduce the danger of a leak. The ammonia compression system, operating at a fairly high back pressure, is generally used for producing the refrigerating effect.

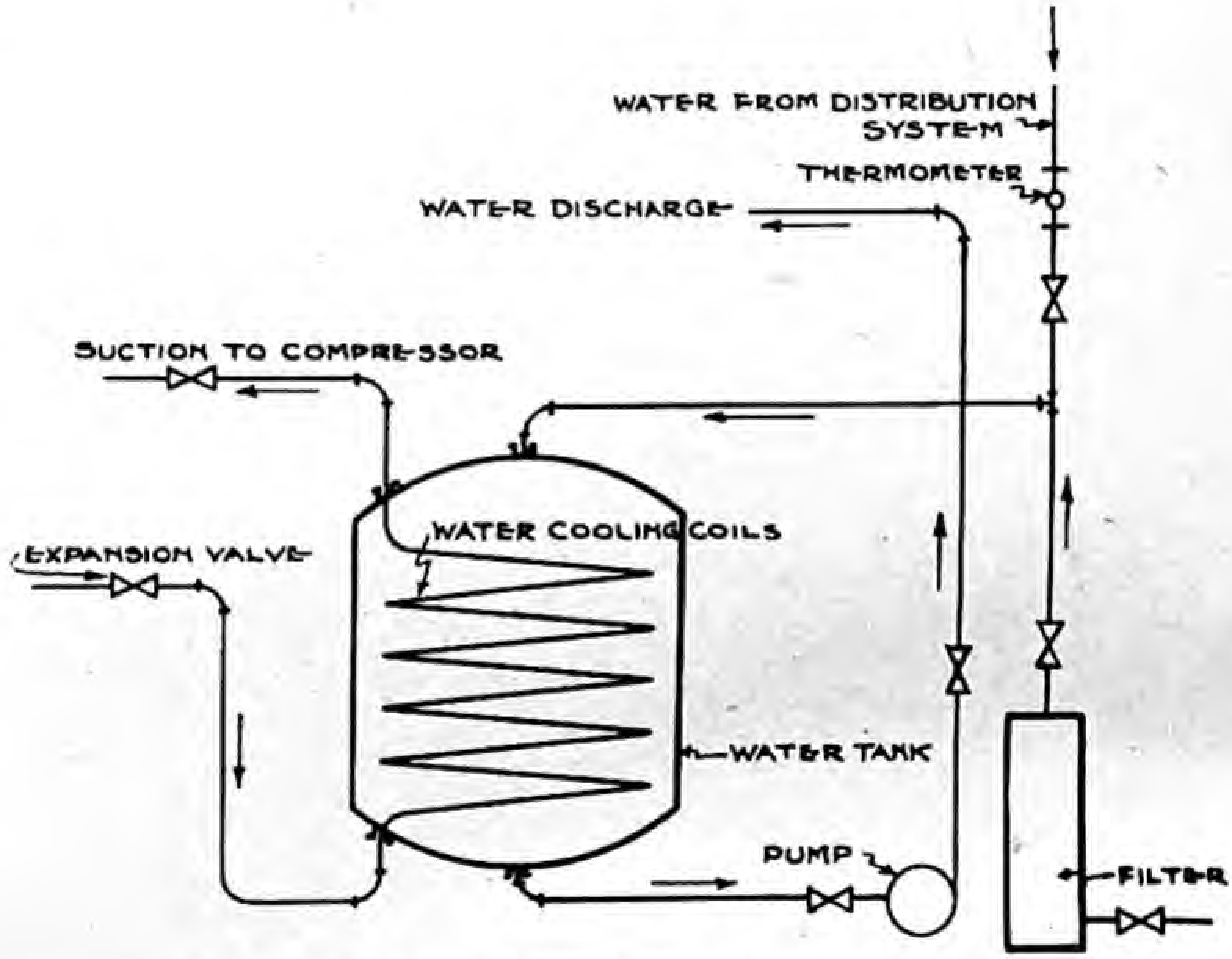


Fig. 155.—Closed Type Water Cooling System.

In the closed system the water is cooled in a closed tank which contains the refrigerating coils. This is shown diagrammatically by Fig. 155. The water, after being cooled, is taken from the bottom of the water cooling tank and is led to the water circulating pump, which discharges the cooled water into the distribution system. After being circulated through the distribution system, it returns to the top of the water cooling tank. The make-up water, or the water that is used

and wasted, is led in through suitable filters to a make-up tank. This water flows from the make-up tank into the main water cooling tank. The distribution system is constructed in the same manner as that for the open type system.

Quantity and Temperature of Water Required.—The amount of water consumed per person depends upon the character of the work, the weather conditions, etc. In general, the quantity consumed per person will vary from 0.10 to 0.40 gals. per-hr. The lowest consumption would be for female employes engaged in work which requires a minimum amount of physical exertion. The figure for the average condition in an industrial plant would be probably 0.25 gals. per person per hr. In steel foundries and other places where the work is strenuous and the temperatures are high, the amount may be equal to as high as 0.30 to 0.40 gals. per-hr. per person.

The most healthful temperature for the cooled water seems to be between 45° and 50° F. Water that is too cold may produce ill effects, while water that is too warm will not quench the thirst. It has been proven from practical experience that water at this temperature is the most acceptable from the various viewpoints and that it is consumed in the larger quantities. The use of water at the temperatures mentioned undoubtedly promotes the more desirable working conditions, resulting in better work or larger volume.

Piping Systems.—As previously indicated, the loops of piping for distributing the water about the building are generally constructed of one-inch to two-inch pipe. The water in these loops of piping is circulated so that the rise of temperature of the water in passing throughout the circuit will not be more than 5° F. In other words, the water will rise from approximately 45° to 50° F. in passing through the circuit. This supplies water to all of the fountains at a desirable temperature. It has been found in practice that the water will remain clear and sparkling if the velocity in the water distributing system is kept below 180 ft.p.m. This fact, together with the length of the loop and the quantity of water consumed, determines the size of the pipe to be used in making the distributing system. The size of pipes that are generally used in the construction of the water distributing system, the capacity of such pipes in order to limit the velocity of the water to three feet per second, or 180 ft.p.m., and the maximum permissible length of same are shown in Table 79.

The velocity of the flow of water through a pipe may be determined by dividing the quantity charged in a given length of time by the area of the pipe. This may also be expressed in a formula as follows:

TABLE 80.—DIMENSIONS OF STANDARD PIPE.

Diameter.			Thickness, in.	Circumference		Transverse areas.			Length of pipe per sq. ft. of		Length of pipe containing one cu. ft., ft.	Wt. per ft. of length, lbs.	No. of thr'ds per in. of screw.	Contents in gals. per ft. of length,	Wt. of water per ft. of length, lbs.
Nominal, internal, in.	Actual external, in.	Actual internal, in.		External, in.	Internal, in.	External, sq. in.	Internal, sq. in.	Metal, sq. in.	External surface, ft.	Internal surface, ft.					
1/8	.405	.27	.068	1.272	.848	.129	.0573	.0717	9.44	14.15	2513.	.241	27	.0006	.005
1/4	.54	.364	.088	1.696	1.144	.229	.1041	.1249	7.075	10.49	1383.3	.42	18	.0026	.021
3/8	.675	.494	.091	2.121	1.552	.358	.1917	.1663	5.657	7.73	751.2	.559	18	.0057	.047
1/2	.84	.623	.109	2.639	1.957	.554	.3048	.2492	4.547	6.13	472.4	.837	14	.0102	.085
3/4	1.05	.824	.113	3.299	2.589	.866	.5333	.3327	3.637	4.635	270.	1.115	14	.0230	.190
1	1.315	1.048	.134	4.131	3.292	1.358	.8626	.4954	2.904	3.645	166.9	1.668	11½	.0408	.349
1¼	1.66	1.38	.14	5.215	4.335	2.164	1.496	.668	2.301	2.768	96.25	2.244	11½	.0638	.527
1½	1.9	1.611	.145	5.969	5.061	2.835	2.038	.797	2.01	2.371	70.66	2.678	11½	.0918	.760
2	2.375	2.067	.154	7.461	6.494	4.43	3.356	1.074	1.608	1.848	42.91	3.609	11½	.1632	1.356
2½	2.875	2.468	.204	9.032	7.753	6.492	4.784	1.708	1.328	1.547	30.1	5.739	8	.2550	2.116
3	3.5	3.067	.217	10.996	9.636	9.621	7.388	2.243	1.091	1.245	19.5	7.536	8	.3673	3.049
3½	4.	3.548	.226	12.566	11.146	12.566	9.887	2.679	.955	1.077	14.57	9.001	8	.4998	4.155
4	4.5	4.026	.237	14.137	12.648	15.904	12.73	3.174	.849	.949	11.31	10.665	8	.6528	5.405
4½	5.	4.508	.246	15.708	14.162	19.635	15.961	3.674	.764	.848	9.02	12.34	8	.8263	6.851
5	5.563	5.045	.259	17.477	15.849	24.306	19.99	4.316	.687	.757	7.2	14.502	8	1.020	8.500
6	6.625	6.065	.28	20.813	19.054	34.472	28.888	5.584	.577	.63	4.98	18.762	8	1.469	12.312
7	7.625	7.023	.301	23.955	22.063	45.664	38.738	6.926	.501	.544	3.72	23.271	8	1.999	16.662
8	8.625	7.982	.322	27.096	25.076	58.426	50.04	8.386	.443	.478	2.88	28.177	8	2.611	21.750
9	9.625	8.937	.344	30.238	28.076	72.76	62.73	10.03	.397	.427	2.29	33.701	8	3.300	27.500
10	10.75	10.019	.366	33.772	31.477	90.763	78.839	11.924	.355	.382	1.82	40.065	8	4.081	34.000
11	12.	11.25	.375	37.690	35.343	113.098	99.402	13.696	.318	.339	1.456	45.95	8	......	......
12	12.75	12	.375	40.055	37.7	127.677	113.098	14.579	.299	.319	1.27	48.985	8	......	......
..	14.	13.25	.375	42.982	41.626	153.938	137.88	16.051	.273	.288	1.04	53.922	8	......	......
..	15.	14.25	.375	47.124	44.768	176.715	159.485	17.23	.255	.268	.903	57.893	8	......	......
..	16.	15.25	.375	50.265	47.909	201.062	182.655	18.407	.239	.250	.788	61.77	8	......	......
..	18.	17.25	.375	56.549	54.192	254.47	233.706	20.764	.212	.221	.616	69.66	8	......	......
..	20.	19.25	.375	62.832	60.476	314.16	291.04	23.12	.191	.198	.495	77.57	..	......	......
..	22.	21.25	.375	69.115	66.759	380.134	354.657	25.477	.174	.179	.406	85.47	..	......	......
..	24.	23.25	.375	75.398	73.042	452.39	424.558	27.832	.159	.164	.339	93.37	..	......	......

TABLE 81.—DIMENSIONS OF EXTRA AND DOUBLE EXTRA STRONG PIPE.

Diameter.			Thickness, in.	Nearest wire gauge, No.	Circumference.		Transverse area.			Length of pipe in ft. per sq. ft. of		Nominal wt. in lbs. per ft.
Nominal internal, in.	Actual external, in.	Actual internal, in.			External, in.	Internal, in.	External, sq. in.	Internal, sq. in.	Metal, sq. in.	External surface.	Internal surface.	
1/8	.405	.205	.1	12½	1.272	.644	.129	.033	.086	9.433	18.632	.29
1/4	.54	.294	.123	11	1.696	.924	.229	.068	.161	7.075	12.986	.54
3/8	.675	.421	.127	10¼	2.121	1.323	.358	.139	.219	5.657	9.07	.74
1/2	.84	.542	.149	9	2.639	1.703	.554	.231	.323	4.547	7.046	1.09
3/4	1.05	.736	.157	8½	3.299	2.312	.866	.452	.414	3.637	5.109	1.39
1	1.315	.951	.182	7	4.131	2.988	1.358	.71	.648	2.904	4.016	2.17
1¼	1.66	1.272	.194	6½	5.215	3.996	2.164	1.271	.893	2.301	3.003	3.
1½	1.9	1.494	.203	6	5.969	4.694	2.835	1.753	1.082	2.01	2.556	3.63
2	2.375	1.933	.221	5	7.461	6.073	4.43	2.935	1.495	1.608	1.975	5.02
2½	2.875	2.315	.28	2	9.032	7.273	6.492	4.209	2.283	1.328	1.649	7.67
3	3.5	2.892	.304	1	10.996	9.085	9.621	6.569	3.052	1.091	1.328	10.25
3½	4.	3.358	.321	0	12.566	10.549	12.566	8.856	3.71	.955	1.137	12.47
4	4.5	3.818	.341	0	14.137	11.995	15.904	11.449	4.455	.849	1.	14.97
5	5.563	4.813	.375	00	17.477	15.120	24.306	18.193	6.12	.687	.793	20.54
6	6.625	5.75	.437	000	20.813	18.064	34.472	25.967	8.505	.577	.664	28.58

DOUBLE EXTRA STRONG PIPE—Table of Standard Dimensions.

1/2	.84	.244	.298	1	2.639	.766	.554	.047	.507	4.547	15.667	1.7
3/4	1.05	.422	.314	1	3.299	1.326	.866	.139	.727	3.637	9.049	2.44
1	1.315	.587	.364	00	4.131	1.844	1.358	.271	1.087	2.904	6.508	3.65
1¼	1.66	.885	.388	00	5.215	2.78	2.164	.615	1.549	2.304	4.317	5.2
1½	1.9	1.088	.406	000	5.969	3.418	2.835	.93	1.905	2.01	3.511	6.4
2	2.375	1.491	.442	0000	7.461	4.684	4.43	1.744	2.686	1.608	2.561	9.02
2½	2.875	1.755	.560	9/16−	9.032	5.513	6.492	2.419	4.073	1.328	2.176	13.68
3	3.5	2.284	.608	5/8−	10.996	7.175	9.621	4.097	5.524	1.091	1.672	18.56
3½	4.	2.716	.642	5/8+	12.566	8.533	12.566	5.794	6.772	.955	1.406	22.75
4	4.5	3.136	.682	11/16−	14.137	9.852	15.904	7.724	8.18	.849	1.217	27.48
5	5.563	4.063	.75	3/4	17.477	12.764	24.306	12.965	11.34	.687	.940	38.12
6	6.625	4.875	.875	7/8	20.813	15.315	34.472	18.666	15.896	.577	.784	53.11

$$V = \frac{G.\,P.\,M.}{2.448\,D^2}$$

where V = velocity of the flow in ft. per sec.
G. P. M. = gal. per min. discharged by pipe
D = actual internal diameter of the pipe, inches
2.448 = a constant

The dimensions of standard heavy and extra heavy steel or wrought iron pipe are given in Tables 80 and 81.

TABLE 79.—DRINKING WATER PIPING SYSTEMS.

Pipe size	Gals. per min.	Maximum length of circuit in feet
½	2.8	600
¾	5.0	1200
1	8.1	1700
1¼	14.0	2600
1½	19.0	3200
2	31.5	4800
2½	44.5	5700

Refrigeration Requirements.—Refrigeration is required for cooling the make-up water or the water that is used and wasted, to offset the heat loss through the insulation on the water distributing system, to offset the heat generated by the friction of the water in the system,

TABLE 82.—HEAT TRANSMISSION OF NONPAREIL CORK COVERING FOR COLD PIPES.
Btu. per lin. ft. per deg. of temp. diff. in 24 hours.

Pipe Size inches	Ice Water thickness	Std. Brine thickness	Special thickness
¼	2.42	2.12	1.64
⅜	2.74	2.36	1.84
½	3.17	2.71	2.13
¾	3.37	2.87	2.39
1	3.64	3.11	2.43
1¼	4.15	3.22	2.70
1½	4.50	3.32	2.84
2	5.20	3.80	3.20
2½	6.17	4.21	3.75
3	6.55	4.49	4.21
3½	7.00	4.78	4.00
4	7.46	5.55	4.49
4½	8.05	5.26	4.28
5	8.90	6.05	4.78
6	9.53	6.85	4.95
7	11.00	7.68	5.71
8	12.10	7.88	6.20
9	13.15	8.15	6.76
10	13.70	9.02	7.33
12	18.00	10.40	8.36
14		11.30	9.00
16		12.60	10.00

and to make up for other losses such as the absorption of heat through the water cooling tanks, etc. The refrigeration required for the cooling of the make-up water will generally consist of cooling the water from the temperature of the water in the city supply mains to the tempera-

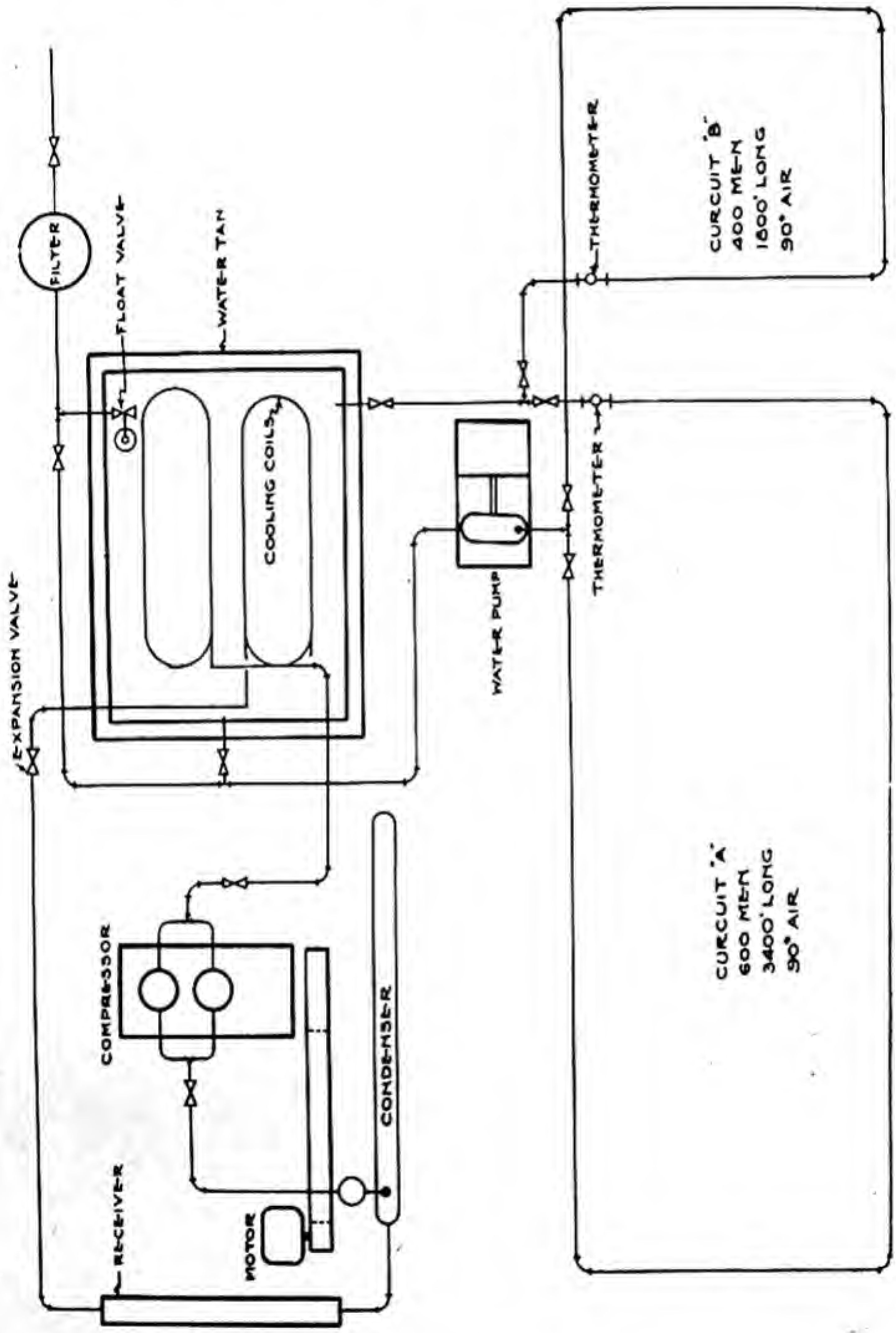


Fig. 156.—Layout of Drinking Water Cooling System.

ture of the water leaving the water cooler. The temperature of the water coming from the city supply mains in the summer time will vary from 60° to 75° F. The temperature of the water leaving the cooler

should be approximately 45° F., and it should return at a temperature near 50° F.

The heat absorbed by the water distributing system is a large part of the total refrigeration load on the plant. The quantity of heat absorbed will depend upon the relative size of the pipe, the mean temperature difference between the air and the water, and the thickness and kind of insulation used upon the pipe. The heat transmission for cork covering in Btu. per lineal foot per degree of temperature difference per 24 hours for different sizes of pipe is shown in Table 82.

This table gives the transmission of Nonpareil cork covering of ice water, standard brine, and special brine thickness. The ice water thickness is generally used upon the pipes for drinking water systems. The heat loss through the insulation on the water coolers may be estimated by the methods which were outlined in Chapter VIII. The refrigeration required to offset the heat generated by the friction of the system may be allowed for at the rate of one Btu. for 778 ft-lbs., which is the mechanical equivalent of heat.

Figure 156 shows a layout of a typical water cooling plant such as would be be installed in a factory. The water from the city supply mains, after passing through a water meter and filter, enters a water cooling tank. The water circulating pump takes the water from the cooling tank and discharges it into the two circuits. A small vertical single-acting ammonia compressor with suitable auxiliary equipment produces the necessary refrigerating effect. The refrigeration required for the two circuits may be estimated in the following manner, if the circuits have the lengths indicated by Fig. 156 and each man consumes 0.35 gallon of water per hour.

CALCULATIONS FOR CIRCUIT A.

Length of lines	= 3,400 ft.
Number of workmen	= 600
Water consumption, 600 × 0.35	= 210 gals. per hr.
Approximate size of lines, from Table 79	= $1\frac{1}{2}$ in.
Heat absorbed by line, $3{,}400 \times \dfrac{4.50}{24} \times (90\text{-}47.5)$	= 27,093 Btu. per hr.
Weight of water to absorb heat, $27093 \div (50 - 45)$	= 5,418 lbs. per hr.
Gallons per hr. $6{,}340 \div 8.33$	= 650
Gallons per min. $= \dfrac{650 + 210}{60}$	= 14.3
Velocity of water $= \dfrac{14.3}{2.448 \times 1.61 \times 1.61}$	= 2.25 ft. per sec.
	= 135 ft. per min.

CALCULATIONS FOR CIRCUIT B.

Length of line	= 1,800
Number of workmen	= 400
Water consumption 400 × 0.35	= 140 gals. per hr.
Size of line	= 1 in.
Heat absorbed by line $1{,}800 \times \frac{3.64}{24} \times (90\text{-}47.5)$	= 11,602 Btu. per hr.
Weight of water to absorb heat, 11,602 ÷ (50-45)	= 2,320 lbs. per hr.
Gallons per hr. 2,320 ÷ 8.33	= 278
Gallons per min. $\frac{140 + 278}{60}$	= 6.96
Velocity of water $\frac{6.96}{2.448 \times 1.048 \times 1.048}$	= 2.59 ft. per sec.
	= 155 ft. per min.

CALCULATION OF MAIN SUPPLY LINE.

Size to supply 1½ in. and 1 in. branches	= 2 in.
Length of line, approximate	= 100 ft.
Amount of water = 14.3 + 6.96	= 21.26 gals.
Velocity of water = $\frac{21.26}{2.448 \times 2.067 \times 2.067}$	= 2.03 ft. per sec.
	= 122 ft. per min.
Heat absorbed = $100 \times \frac{5.20}{24} \times (90\text{-}45)$	= 975 Btu. per hr.
Weight of water to absorb heat = 21.26 × 60 × 8.33	= 10,625
Temperature loss = $\frac{975}{10{,}625}$	= 0.091° F.
Temperature of water from tank, 45° − 0.09°	= 44.91° F.

CALCULATION OF MAIN RETURN LINE.

Size	= 1½ in.
Length of line, approximate	= 75 ft.
Amount of water $\frac{650 + 278}{60}$	= 15.5 gals. per min.
Velocity of water $\frac{15.5}{2.448 \times 1.61 \times 1.61}$	= 2.44 ft. per sec.
	= 146 ft. per min.
Heat absorbed $75 \times \frac{4.50}{24} \times (90\text{-}50)$	= 562 Btu. per hr.
Weight of water, 15.5 × 60 × 8.33	= 7,747
Temperature loss $\frac{562}{7{,}747}$	= 0.072° F.
Temperature of water returned to tank, 50° + 0.072° F.	= 50.07° F.

REFRIGERATION FOR WATER.

Heat to cool supply water $(210 + 140) \times 8.33 \times (75\text{-}44.91)$	$= 87{,}727$ Btu.
Heat to cool return water $(650 + 278) \times 8.33 \times (50.07\text{-}44.9)$	$= 39{,}890$
Total refrigeration	127,617
Refrigeration capacity, $127{,}617 \div 12{,}000$	$= 10.6$ tons

Water Cooling for Industrial Purposes.—In addition to cooling water for drinking purposes, mechanical refrigeration is used quite extensively at present for cooling water for various industrial plants. Water at a low temperature is now a necessity in such plants as rubber factories, bakeries, wax paper factories, die works, manufacture of chemicals, explosives, etc. In nearly all of these applications of mechanical refrigeration the cold water acts merely as a carrier of refrigeration. This is due to the nature of the apparatus in which the cooling is to be done, and also to the nature of the process under consideration. The refrigerating requirements, therefore, will depend upon the nature of the process, the amount of material being cooled, etc.

The water may be cooled by any of the different types of coolers. These generally consist of a metallic container for the refrigerant. The temperature of the boiling refrigerant is maintained a few degrees below the temperature of the water. The heat then flows by natural tendency from the water into the refrigerant, causing it to be evaporated directly. The type of cooler that is used upon the larger installations consists of a series of vertical direct-expansion pipe coils, over which the water is showered. This cooler has the advantage of being safer to operate, in that a layer of ice forms on the coil surface and only a reduction of capacity will be encountered. Due to the fact that the velocity of the water over the pipe coil surface is appreciable, the heat transmission is good.

Also, in the large installations in which the temperature of the refrigerant isn't very far below the freezing temperature of the water, the double pipe water cooler may be used to advantage. Of course, if the refrigerant temperature is allowed to become too low, the water may freeze in the pipes, causing them to burst. The velocity of the water passing through the double pipe cooler is quite large, which means that the heat transmission is very good. Thus, a small amount of heat transmitting surface is required. The double pipe cooler has the additional advantage of delivering the cooled water to the plant under pressure.

In the smaller installations, the submerged direct-expansion coil water cooler may be used. It is evident also that in any of these water

coolers, that instead of using a liquefiable fluid as the refrigerant, cold brine may be used.

General Refrigeration Requirements.—Since mechanical refrigeration is used for cooling purposes in very many industrial plants, it is evident that the refrigeration requirements for a plant will depend upon the different kinds of work being done, the kinds of materials produced, methods of production, etc. As previously indicated, the refrigeration requirements will depend upon several factors, such as the refrigeration required to offset the heat generated in the room by motors, lights, heating, kettles, men; refrigeration to absorb the heat transmitted through the walls and insulation; refrigeration to cool the air required for ventilation purposes; refrigeration required to cool the material stored, or products of manufacture. In many commercial applications of refrigeration, there are changes of state from liquid to solid, from solid to liquid; from liquid to vapor, and vapor to liquid. The heats evolved during these different processes may be termed the latent heats of fusion, or evaporation, fermentation, heats of composition, crystallization, etc.

Table 3 of Chapter II has been compiled, from various sources, to show the characteristics of various metals, organic compounds, inorganic compounds, etc., which may require refrigeration in industrial plants. The first column of this table gives the names of the elements or compounds. The second column gives the specific gravity with the specific gravity of water being equal to one. The third column gives the temperature range or the temperature in degrees Fahrenheit for the corresponding specific heats. The fourth column gives the values of the specific heat in Btu. per pound. The fifth column gives the melting temperature in degrees Fahrenheit. The sixth column gives the latent heat of fusion in Btu. per pound. The seventh column gives the boiling temperature in degrees Fahrenheit. The last column gives the latent heat of vaporization in Btu. per pound of material. In some cases, the material to be given consideration will not be listed in Table 3, Chapter II, in which case their characteristics may be determined by referring to tables which appear in various well known handbooks.

QUESTIONS ON CHAPTER XIII.

1. In the consideration of refrigeration requirements for creamery and dairy plants, what are the principal parts which require refrigeration?

2. It is desired to cool 2,000 gals. of milk per hr. from 80° to 40° F., and to cool 4,000 gals. of cream per hr. from 70° to 40° F. What is the

refrigeration power required when this work is done on the hourly basis? What is the refrigeration power when the refrigeration is performed on the 24-hr. basis?

3. Ice cream has a swell of 72 per cent, a specific gravity of 1.07, and is cooled from the temperature of the mix at 60° F. to the hardened cream at a temperature of 5° F. Find the actual refrigeration required to cool, freeze, and harden 2,000 gals. per day of 12 hrs.

4. If the losses about the plant amount to 100 per cent of the actual work for cooling and freezing the cream, what would be the actual refrigeration capacity to freeze and harden the cream described in Problem No. 3?

5. If 15 lbs. of ice per gal. are required for packing and shipping purposes, and water is available at 70° F. for ice making purposes, find the total daily tonnage capacity of the above plant when it is desired to freeze and harden 2,000 gals. of ice cream, to cover the various losses, and to make the necessary ice.

6. If one-half the latent heat of fusion and the heat required for cooling before freezing is removed in the ice cream freezer, find the quantity of brine to be circulated in 8 hours to freeze the above 2,000 gals. of ice cream, when the brine is warmed 2° F. in passing through the freezers.

7. A drinking water system consisting of four loops of distributing piping, each 3,000 ft. long and each supplying 500 men with drinking water, is to be installed in a factory where the temperature during the hotter summer months is 90° F. Find the refrigeration capacity of the plant to be installed to produce the cooling effect equal to the heat leakage through the insulation on the cold water pipes and that required for cooling the water that is used.

8. Determine the compressor cylinder size for the refrigeration plant described in Problem No. 7, when a slow-speed horizontal double-acting ammonia compressor is used which operates between the suction pressure of 40 lbs. per sq. in. gauge and the condensing pressure of 185 lbs., and which has a stroke equal to twice the diameter of the piston.

9. Determine the amount of direct-expansion coils to be used in the water cooler for the above water cooling plant, when the overflow pipe coils are used and when the submerged pipe coils are used. Also, what would be the capacity of the water circulating pump in gallons per minute?

10. Quenching oil is to be cooled from 95° to 60° F. at the rate of 6,000 gals. per hr. The specific gravity of the oil is 0.900 and the specific heat is 0.45. What would be the refrigerating capacity of the machine to produce the desired cooling effect if the ammonia is allowed to absorb all of the heat required to cool the oil from 95° to 60° F.?

CHAPTER XIV.

COLD STORAGE BUILDINGS.

Benefits of Cold Storage.—Many different articles are stored at present at low temperatures. The purpose of this cold storing is chiefly the preservation or the prevention of decay. Food commodities are probably the most important materials which are cold stored. Many food products are produced during certain seasons of the year and it is evident that there would be great waste from decomposition before they could reach the ultimate consumer if use were not made of refrigeration.

Quite similarly, the use of refrigeration lengthens the period of consumption of various food products. It is now possible to have a variety of the different foods throughout the various seasons of the year. This improves the health of mankind in general. Also, by the use of refrigeration, food may be transported from the points of production to the more thickly settled centers of population. It is possible by means of refrigeration to ship food products across the continent or across the seas. Due to the fact that the use of mechanical refrigeration effects the conservation of the food supply, as well as the efficient transportation of same, prices of such food products are lowered and equalized.

The present development of the cold storage industry as a public benefactor has depended almost entirely upon the introduction and the utilization of refrigeration produced by mechanical means whereby temperatures are assured ranging from $-15°$ F. to $40°$ F. The ease with which these various temperatures are maintained by mechanical means in a properly constructed and insulated cold storage warehouse is surely one of the modern types of industrial science.

Articles Placed in Cold Storage.—As previously indicated, numerous articles are stored at various low temperatures. Among the food products stored, the following are the most important: Meats, poultry, game, fruits, vegetables, fish, oysters, eggs, dairy products. It will no doubt be interesting to note the extent of the use of low temperatures

for storing commercial commodities. As an indication of the numerous articles which are placed in cold storage warehouses at present the following list, taken from *Ice and Refrigeration Blue Book and Buyers' Guide,* shows that as high as 125 different articles may be found in one of the largest cold storage warehouses:

LIST OF ARTICLES HELD IN COLD STORAGE.

Anchovies	Cucumbers	Lard	Pineapples
Apples	Currants	Laurel Leaves	Plants
Apple Waste	Dates	Leeks	Potatoes
Apricots	Dried Fish	Lemons	Poultry
Asparagus	Dried Fruits	Lettuce	Preserves
Bananas	Dried Meats	Limes	Provisions
Beans	Eggs	Lobsters	Prunes
Beer	Eggplant	Macaroni	Radishes
Berries	Evap. Apples	Maple Sugar	Raisins
Brussels Sprouts	Evap. Peaches	Maple Syrup	Rhubarb
Buckwheat	Fabrics	Meats, Fresh	Rice
Bulbs	Ferns	Melons	Salad Dressing
Butter	Figs	Mushrooms	Sauerkraut
Cabbages	Fish for Bait	Nursery Stock	Sausage Casings
California Fruits	Flour	Nuts	Scallops
Candied Fruits	Flowers	Oils	Shallots
Canned Goods	Food Fish, Fresh	Oleomargarine	Shrimp
Carrots	Fruit Juices	Olives	Silks
Cauliflower	Furniture	Olive Oil	Skins
Caviar	Furs	Onions	Smilax Leaves
Cereals	Game	Oranges	Smoked Fish
Cheese	Garments	Oysters	Smoked Meats
Cherries	Grapes	Parsley	Spinach
Chestnuts	Grape Fruit	Parsnips	Sponges
Cider	Gutta Percha	Peaches	Squash
Citron	Hams	Peanuts	String Beans
Clams	Herbs	Pears	Sweetbreads
Condensed Milk	Holly	Peas	Syrups
Confectionery	Honey	Peppers	Turnips
Crabs	Hops	Pickles	Wines
Cranberries	Horseradish	Pickled Fish	Woolens
Cream	Jellies	Pickled Meats	Yarn

To this list might be added such articles as anatomical parts, vaccine lymph, grains, seeds, roots, hides, leather, canned goods, and many other items.

The proper storage temperatures, specific heats, latent heats of fusion, of some of the common materials placed in cold storage are given in Table 83, Page 472.

General Location of Warehouses.—Since the commercial cold storage warehouse may be classed as a public benefactor, it should be located at a point which is most convenient to the trade. It is evident that consideration should be given to the country producer and shipper, as well as the wholesale and commission merchants in the city in

which the cold storage plant is located. A suitable location for such a cold storage plant would be one that would best serve those who make use of the plant.

It is evident that such a plant should be located near the railroad so that carload shipments may be received or shipped from the plant. It also should be located in the center of its distribution district for the merchants in the city. Ample amount of private switch tracks should be provided so that several cars may be loaded or unloaded at the same time. Of course, the exact amount of trackage to be installed will depend upon the size of the plant, as well as the nature of the work which is being done. Also, in considering the general arrangement of the plant relative to the teaming and railroad facilities, it is evident that the plant should be so laid out that the labor required for handling the material would be reduced to a minimum. Especial attention should be given to the methods of handling the local trade by teams or trucks.

In the event that trucks and wagons are allowed to drive across the sidewalk or street up to the property line of the building, a narrow loading platform may be constructed just on the outside of the building. This arrangement saves a valuable space which would otherwise be occupied in the building. The goods may be loaded from the platform into the rooms, or from the rooms onto the plaform. In the event that the loading and unloading must be done within the boundaries of the property, a practical method is to construct a loading court with a platform on the property so that the loading will not interfere with the traffic on the street. This loading court generally occupies part of the first floor and is near the level of the street. The second floor of the building, of course, may be extended over this loading court, in which case the floor is supported by suitable columns. The space required for such a loading platform and loading court would be from 30 to 40 ft. in depth.

An additional consideration relative to the general location of the cold storage plant is that of handling the material required for the plant. This is especially true of fuel and water. In the event that the plant is driven by means of steam engines, proper facilities for handling and storing fuel should be provided. In a similar manner the water which is required for the operation of the plant, as well as that for the ammonia condensers, should receive attention. It is evident that the providing of a sufficient amount of water for the plant purposes is a question of economic importance. Thus, it is necessary to compare the cost of the water when taken from the city mains with the cost of supplying the water from wells, lakes, rivers, or canals, by means of equipment installed in the plant. Also, it is evident that the relative geographical location of the plant, which in turn affects the temperature of the water, is an important consideration.

From the foregoing it will be observed that the ultimate financial success of an individual cold storage warehouse is dependent upon many factors, and that the whole proposition must be analyzed from the various viewpoints in order to arrive at a successful solution of the problem.

Cold Storage Warehouses.—Modern cold storage warehouses have been built principally in two types of construction. These are the fireproof and the slow-burning types. The permanency of the fireproof construction, the low rate of depreciation, the low maintenance cost, and better sanitary conditions about the building are some of the factors in favor of this type of construction. This construction makes it possible to secure the lowest insurance rates. Similarly, in the larger cities, where the value of the land is high, it is necessary to erect tall buildings, and the fireproof construction lends itself to this necessity.

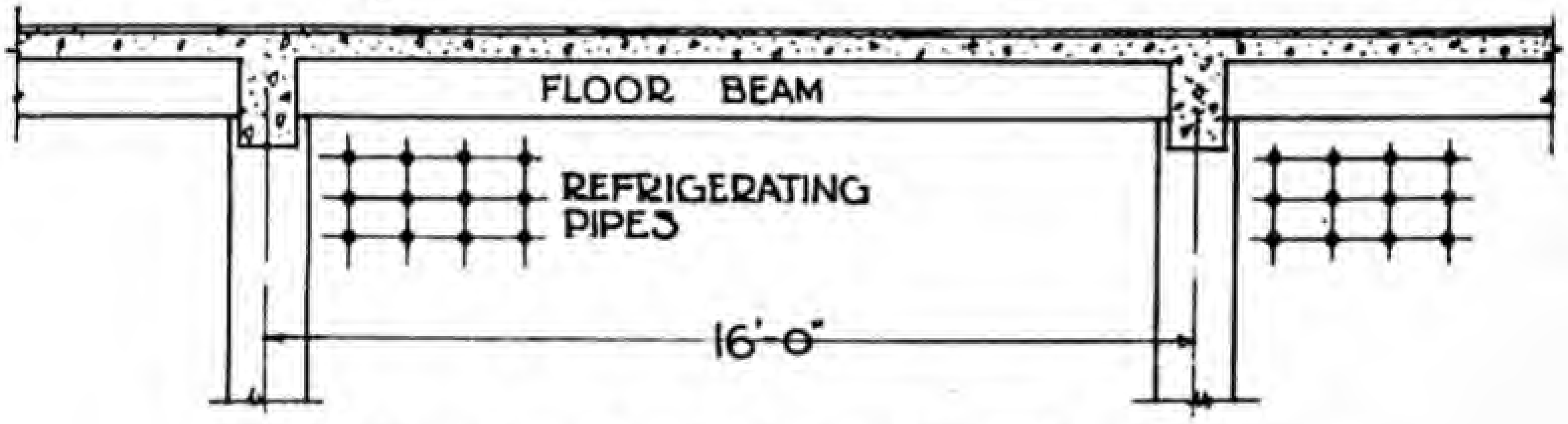


Fig. 157.—Beam and Girder Floor Construction.

On the other hand, when the cost of the construction must be kept down, or in localities where lumber is cheap and when large sizes are available, it may be economical to adopt the slow-burning type of construction. This construction is generally termed "mill construction" and means a slow-burning building which has incombustible walls and roof, and in which the columns supporting the floor are not less than 10 in. by 10 in., with the girders and other principal members of the structure in the same proportion. All structural iron and steel which is used in erecting such a building must be covered with two inches of fireproof material. This type of building must be protected by a suitable automatic sprinkling system. In this respect it will be noted that on account of the use of low temperatures in many of the cold storage rooms it is necessary to install a sprinkling system which is termed "the dry system." In the event that it is necessary to adopt the slow-burning construction, particular attention should be given to the details of the construction of the building, such as the method of supporting the floors, the size of the columns, the thickness of the floors, the protection of the outside walls, etc.

In the fireproof construction reinforced concrete is used principally at present in cold storage buildings which do not exceed ten stories in height. The maximum height of a reinforced concrete building is determined in some manner by the size of the columns to be used in the lower floors. It is evident that much larger columns must be used when constructed of concrete, as compared to columns of the same strength constructed of steel. The maximum diameter of concrete columns in the lower floors seem to be about 30 in., which, with the ordinary floor loading of cold storage buildings, will limit the height of the building to approximately ten stories. Generally, when it is necessary to make the cold storage warehouse more than ten stories high, it is desirable to use steel construction. This type may be from 10 to 20 per cent more expensive than the reinforced concrete type.

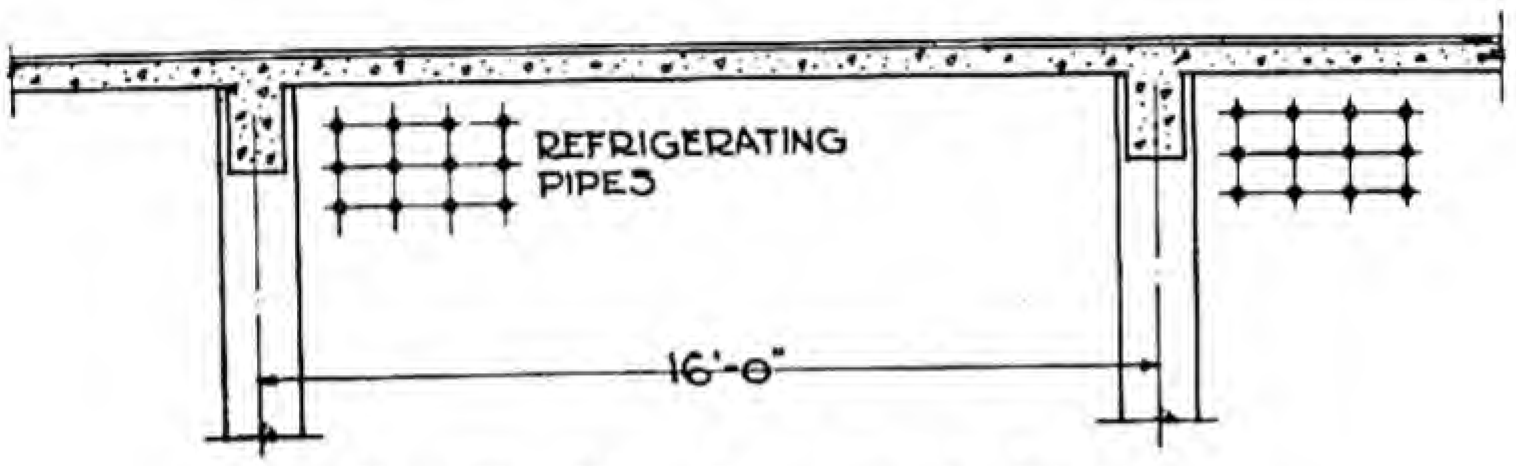


Fig. 158.—Floor Supported by Girders.

Construction of Floors.—Since a large majority of the modern cold storage warehouses are constructed of reinforced concrete, the major consideration of this type of construction will be presented only. In the ultimate analysis it will be noted that the floors of a cold storage warehouse comprise an important element in the building construction and should be designed to meet the special requirements and conditions. Fig. 157 illustrates a type of floor construction which has been used sometimes. This is termed the "beam and girder" construction and is used in order to reduce the thickness of the floor slab to a minimum.

Upon an inspection of this design it will be noted that the refrigerating pipes cannot be placed very near the lower side of the floor slab. This means that there will be considerable waste space at this point, which in turn means that the building must be built higher in proportion to this loss without any gain in the available refrigerated space. An additional objection to this type of construction is that it may interfere with the vigorous circulation of the air about the room, which in turn means that moisture may be precipitated on the ceiling.

Fig. 158 illustrates another type of construction which is similar to the one shown in Fig. 157. In this type of construction the cross-

beams are omitted, the floor being supported on girders which extend from column to column. With this construction the refrigerating pipes may be placed near the ceiling, thereby reducing the necessary height required for the different stories. This type of construction is used where it is necessary that the floor be drained properly in such plants as packing houses, etc.

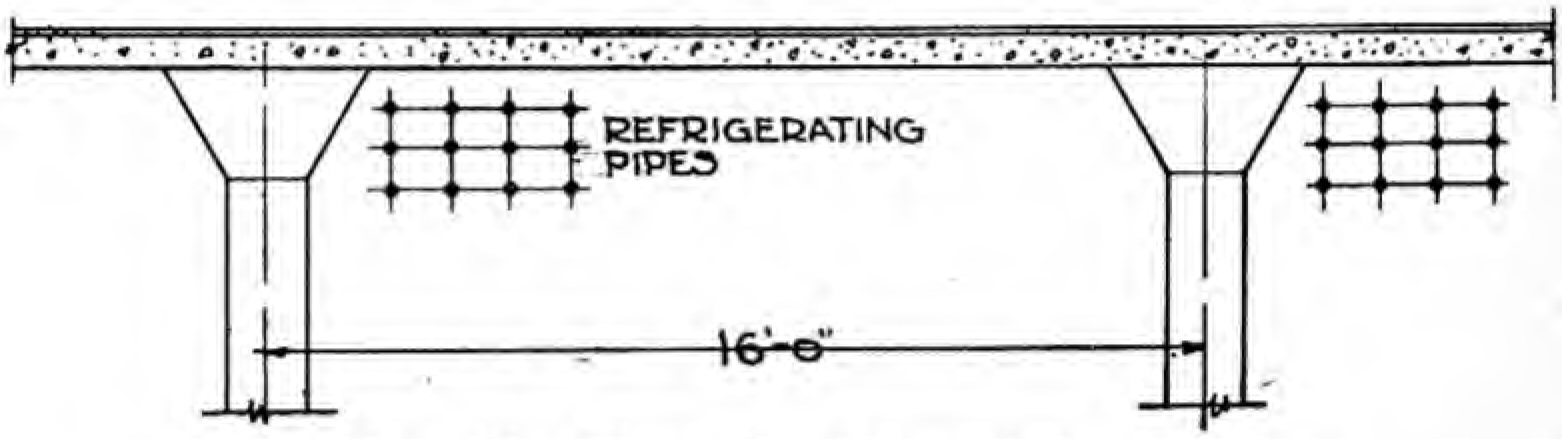


Fig. 159.—Flat Floor Slab Type of Construction.

Fig. 159 illustrates what is known as the "flat floor slab type" of construction. In this the beams and girders are omitted and the floor is made somewhat thicker and is more heavily reinforced. The floor is supported by means of columns which may be placed at suitable centers, varying from 16 to 20 ft. This type of construction adapts itself very well to the construction of cold storage warehouses, and is used extensively at present. The flat ceiling allows a vigorous circulation of air about the room, and the refrigerating pipes may be placed in either direction and close to the ceiling. It is also evident that with this construction the height per story of the building may be reduced to a minimum.

Construction of Walls.—Walls for cold storage warehouses are constructed of many materials at present, such as brick, stone, reinforced concrete, tile, or a combination of these materials. In general, the selection of the type of material to be used in the cold storage wall depends upon the type of the wall. In the higher buildings in which the weights of the walls are supported by the internal skeleton of the building, it is desirable to use as light a material as possible in order to reduce the weight of the building. When the building consists of five stories or less it is generally advisable to use a self-supporting wall to entirely enclose the building. In buildings which are higher than five stories the skeleton construction may be used, in which case the walls are supported by the various floors and columns. These walls are known as curtain walls and are generally constructed of brick or a combination of brick and tile. In either the low or the high building the walls are used simply to enclose the building, the floors and roof being supported by the interior construction.

Since it is the purpose of the walls to retard as much as possible the flow of the heat from the outside to the inside, materials which have a high resistance to the flow of heat should be used. This does not mean, however, that the heat resisting power of the wall should be taken into consideration when estimating the refrigeration requirements, as has been indicated previously, but it means that the wall should be constructed in such a manner as to prevent as much as it is possible to do so the entrance of moisture to the surface of the insulation.

In this respect it will be observed that the use of solid brick or concrete walls will not give such efficient service as some other material which may have a higher moisture resisting power. Thus it will be noted that hollow tile, protected on the outside by a layer of hard brick, would probably prove to be the most economical construction. The hollow tile not only resists the entrance of moisture in a better manner than brick or concrete, but also has a heat resisting power due to the enclosed air spaces. In the erection of hollow tile walls it is generally advisable to plaster the surfaces. The cost of the plastering is not excessive and the appearance of the wall is greatly improved.

Fig. 160 illustrates a cold storage wall which is constructed of hollow tile and hard vitrified brick. This figure shows a self-supporting wall as applied in cold storage warehouse construction. The same construction, however, may be used equally well in the skeleton construction, in which case the wall is known as a curtain wall. Of course, the thickness of the walls is determined by the design of the building, the city ordinances, and the insurance regulations. Nothwithstanding the fact that brick walls are quite porous and will allow very appreciable amounts of moisture to be carried to the insulation, they are used quite extensively for exterior construction at present.

Insulation of Cold Storage Buildings.—As previously mentioned in the chapter pertaining to "Heat Transmission in Insulation and Apparatus," the purpose of the insulation in a cold storage building is to prevent as much as possible the flow of the heat from the outside to the inside of the building. The most efficient method of securing this result would be to entirely enclose the building in an envelope of an efficient insulator, and this is the method pursued in the greater part of the modern cold storage warehouses. The methods adopted for producing this result are shown as follows:

Fig. 160 shows how the insulation on the outside of a building may be made continuous and how it may be connected to the insulation in the floor of the building so as to make a continuous envelope of insulator. The columns in this type of construction are known as split

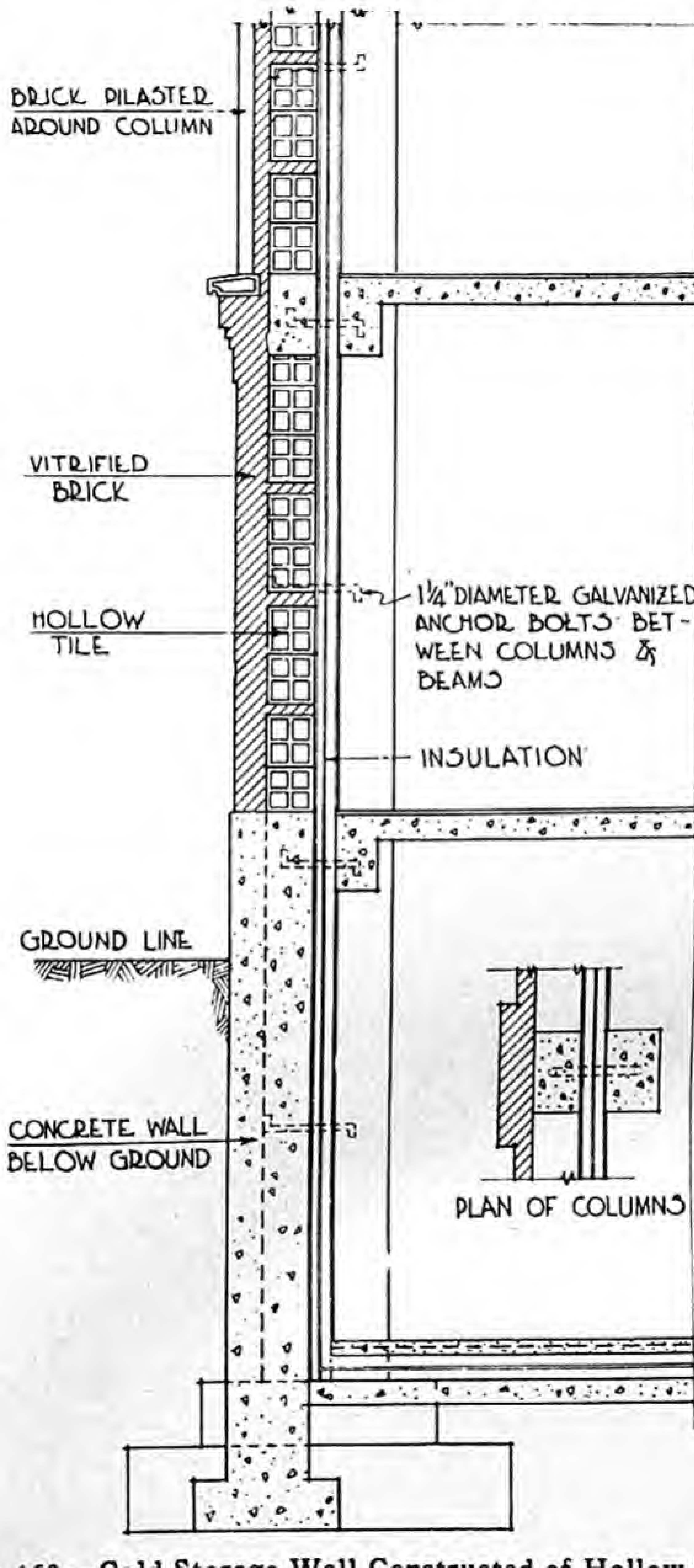


Fig. 160.—Cold Storage Wall Constructed of Hollow Tile and Brick.

columns. The self-supporting wall is anchored securely to the main structure of the building by means of anchor bolts. It is evident that the insulation may be carried in a continuous manner to the roof.

Fig. 161 shows a type of construction which is sometimes used in a smaller building. In this type of construction the load of the floor and roof panels which adjoin the walls are carried by beams which are supported by the walls. In order to reduce the heat transmission to a minimum the insulation is placed around the ends of the beams as shown by Fig. 161.

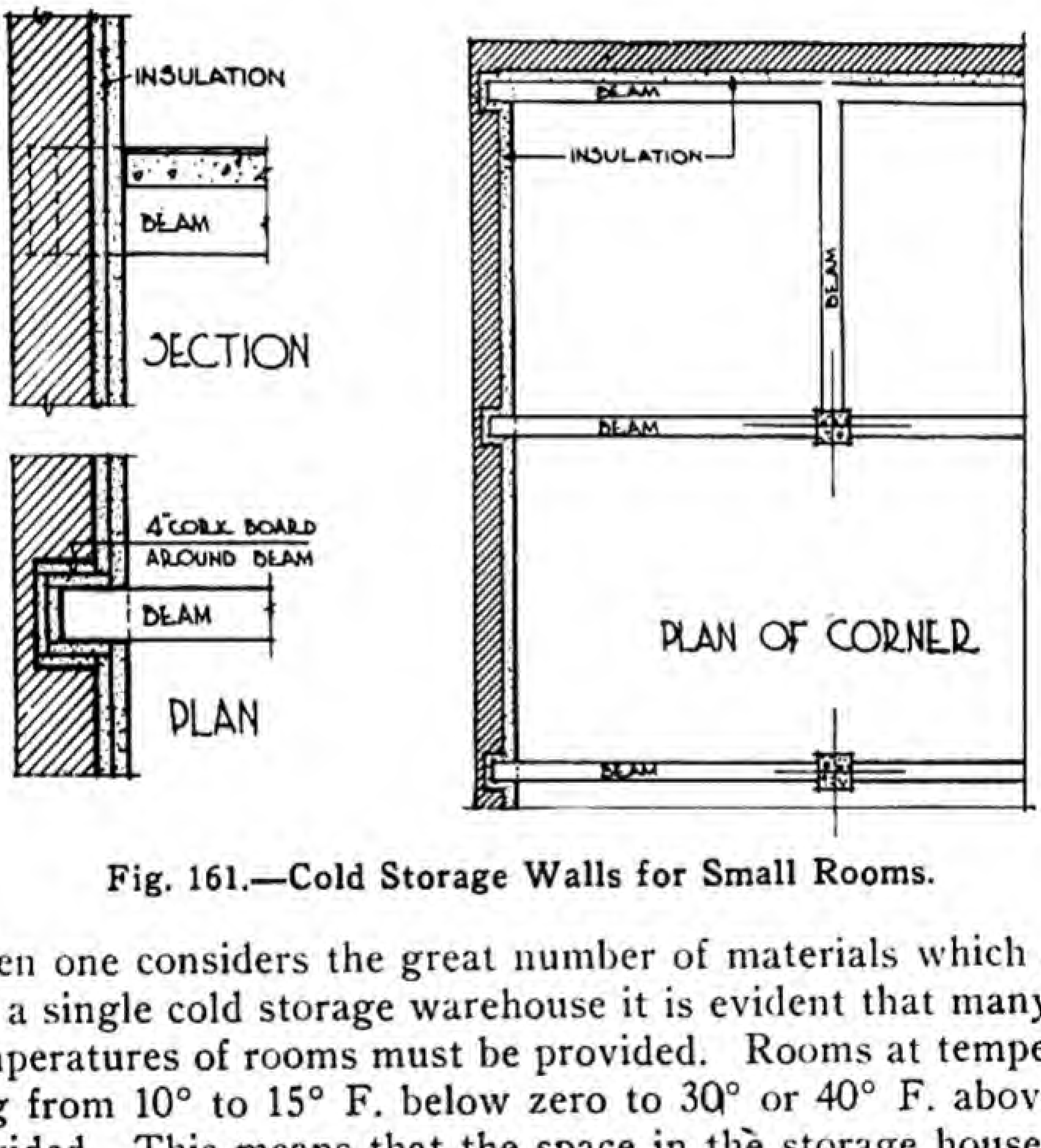


Fig. 161.—Cold Storage Walls for Small Rooms.

When one considers the great number of materials which may be kept in a single cold storage warehouse it is evident that many different temperatures of rooms must be provided. Rooms at temperatures varying from 10° to 15° F. below zero to 30° or 40° F. above, must be provided. This means that the space in the storage house should be divided into what is known as freezer space and cold storge space. It is evident that the division between these rooms must be insulated, since there is quite a difference of temperatures.

One of the most economical and efficient divisions of space consists of dividing the building into sections by means of a vertical insulated partition wall. A division of this kind is shown by Figs. 162, 163 and 164. The building illustrated in these figures has been divided into a cold storage section, an egg section, and a freezer section. The dividing partition should be insulated and should extend from one side of the building to the other and from the floor to the top of the

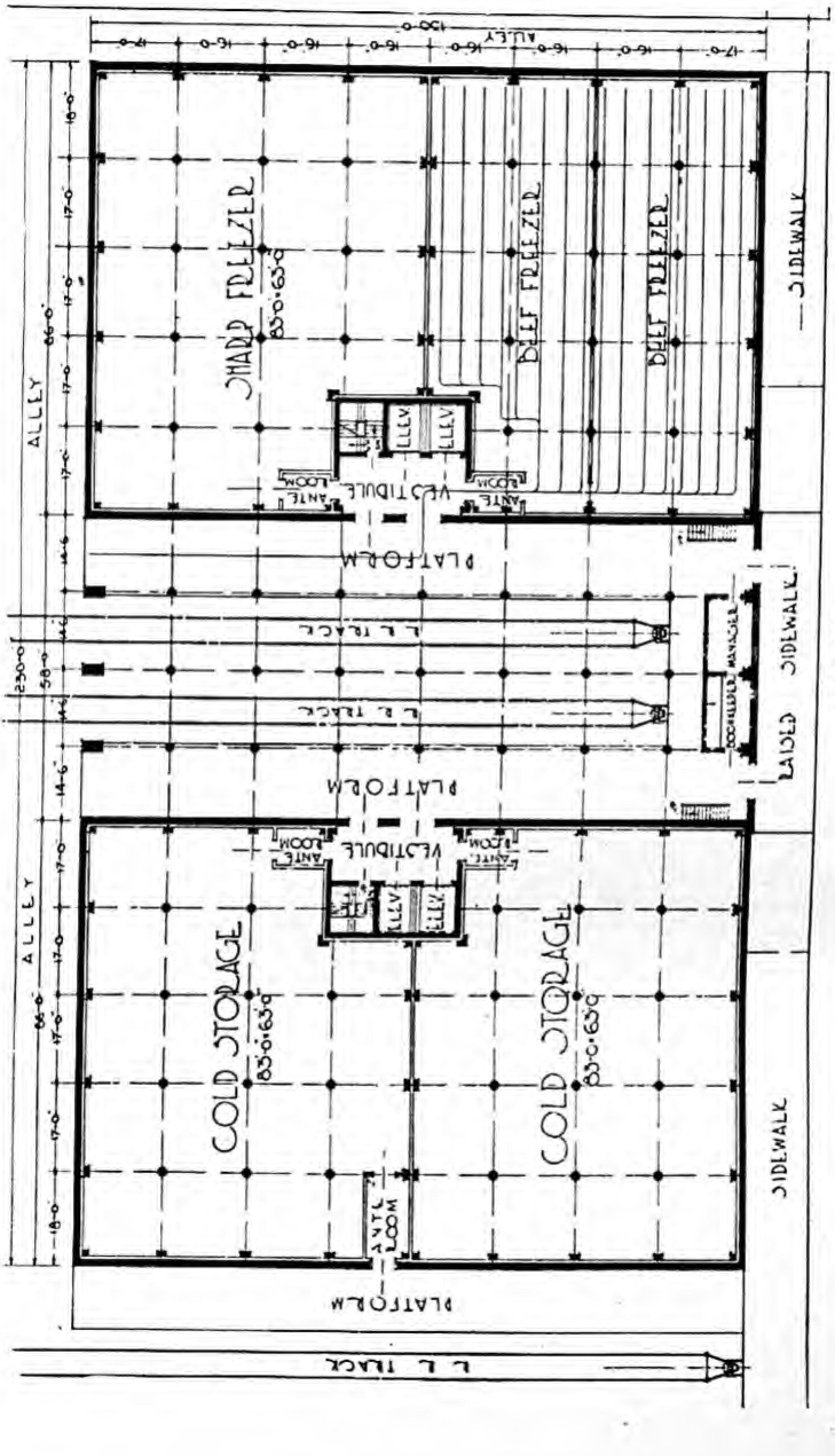


Fig. 162.—First Floor Plan of Merchants' Cold Storage & Warehouse Co. Plant, Chicago, Ill.

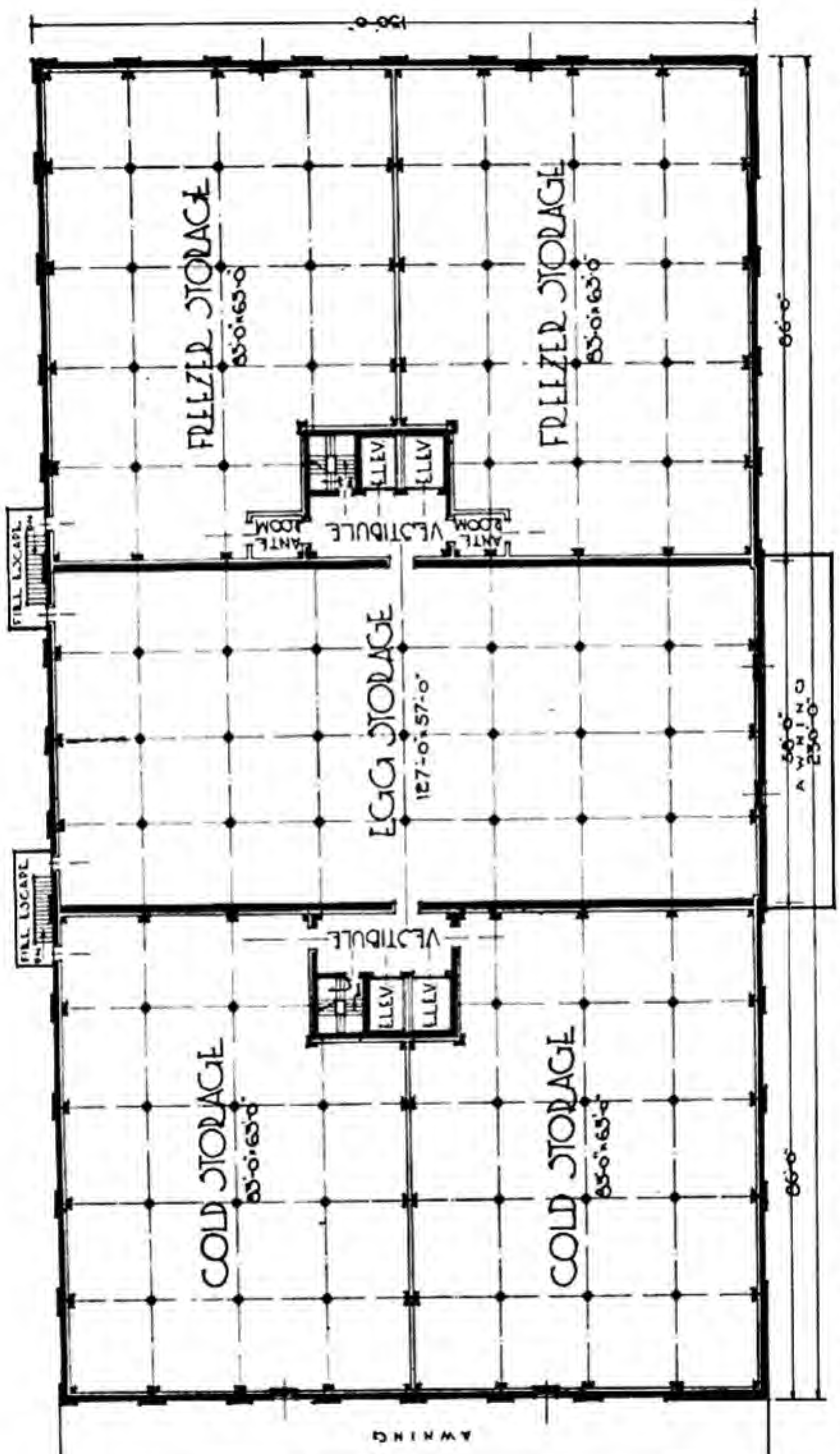


Fig. 163.—Typical Floor Plan of Merchants' Cold Storage & Warehouse Co. Plant, Chicago, Ill.

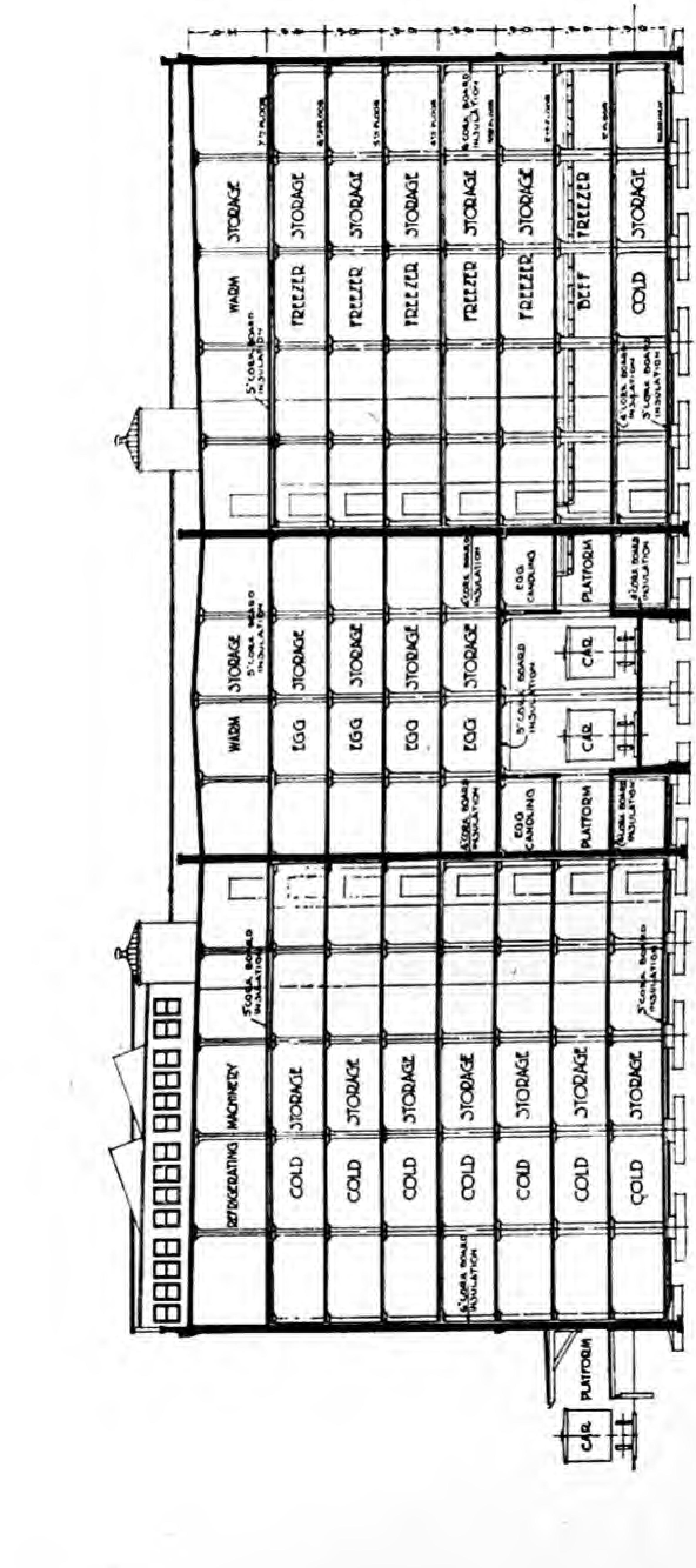


Fig. 164.—Elevation of Merchants' Cold Storage & Warehouse Co. Plant.

building. In order to do this partitions constructed of solid insulating materials, split columns and split girders must be used. This is shown by Fig. 165. In exceptionally tall buildings it is sometimes advisable to make a horizontal division of refrigerated space by application of insulation to the ceiling of every third or fourth story. This makes the operation of the cold storage house more flexible. The insulation should always be applied to the ceiling of the room, and should not, under any circumstances, be placed upon the floor of the room above. Fig. 166 shows how insulation may be applied to a concrete ceiling.

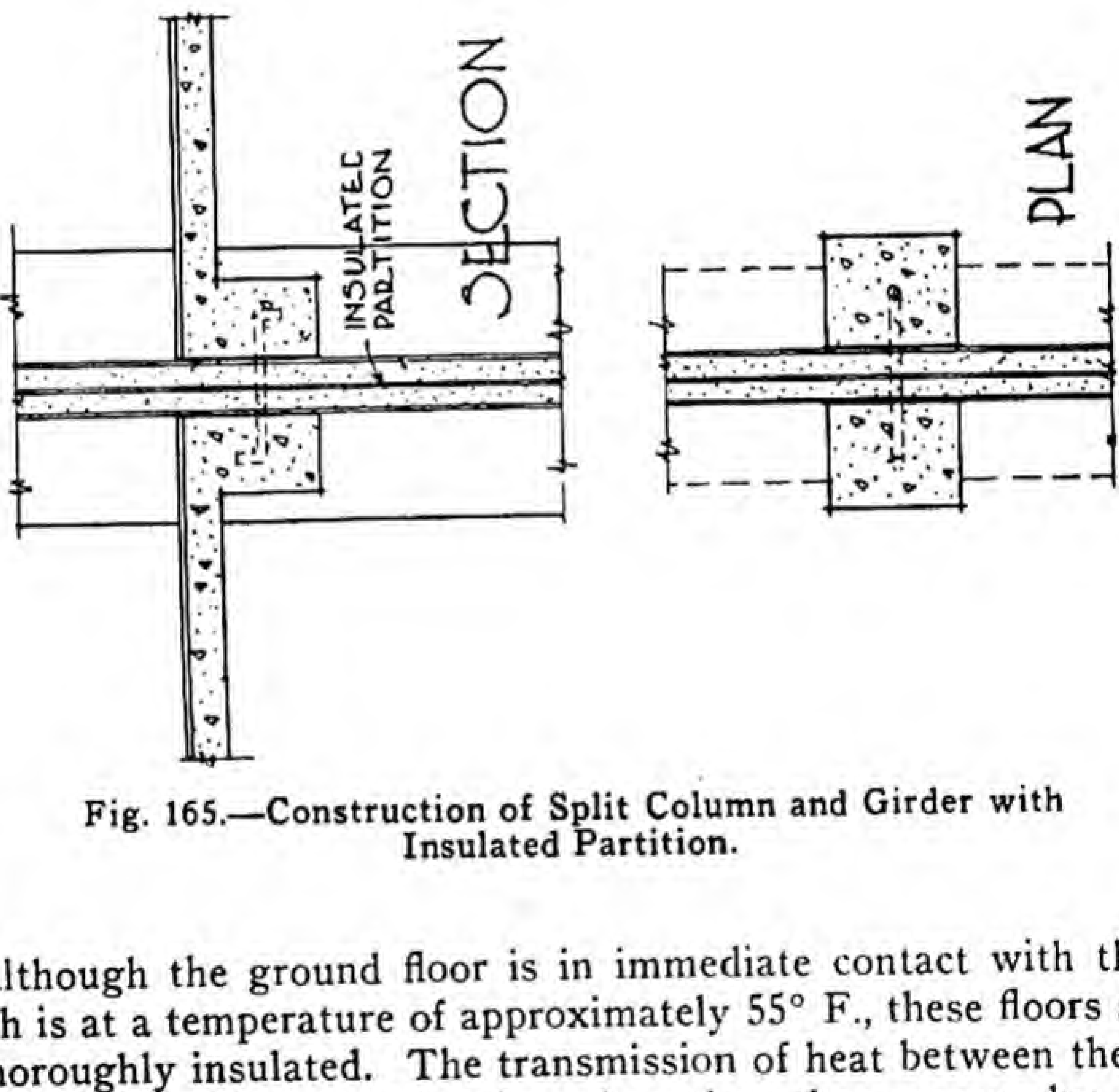


Fig. 165.—Construction of Split Column and Girder with Insulated Partition.

Although the ground floor is in immediate contact with the soil which is at a temperature of approximately 55° F., these floors should be thoroughly insulated. The transmission of heat between the room and the ground is somewhat less than the other rooms, due to the smaller temperature difference, but since there is a temperature difference there is a flow of heat, so it is advisable to insulate the floors. An efficient method of insulating the floors is shown by Fig. 167. This consists of putting down the corkboard insulation upon a concrete floor, which is, in turn, supported by cinders. In the event that the ground contains much water, suitable provision for drainage should be provided. Of course, a concrete wearing surface should be placed immediately on top of the corkboard. Rooms which have an excessively low temperature, such as freezer rooms, should never be placed

upon the ground floor. This is due to the fact that the ground may become frozen, which may affect the foundation of the building.

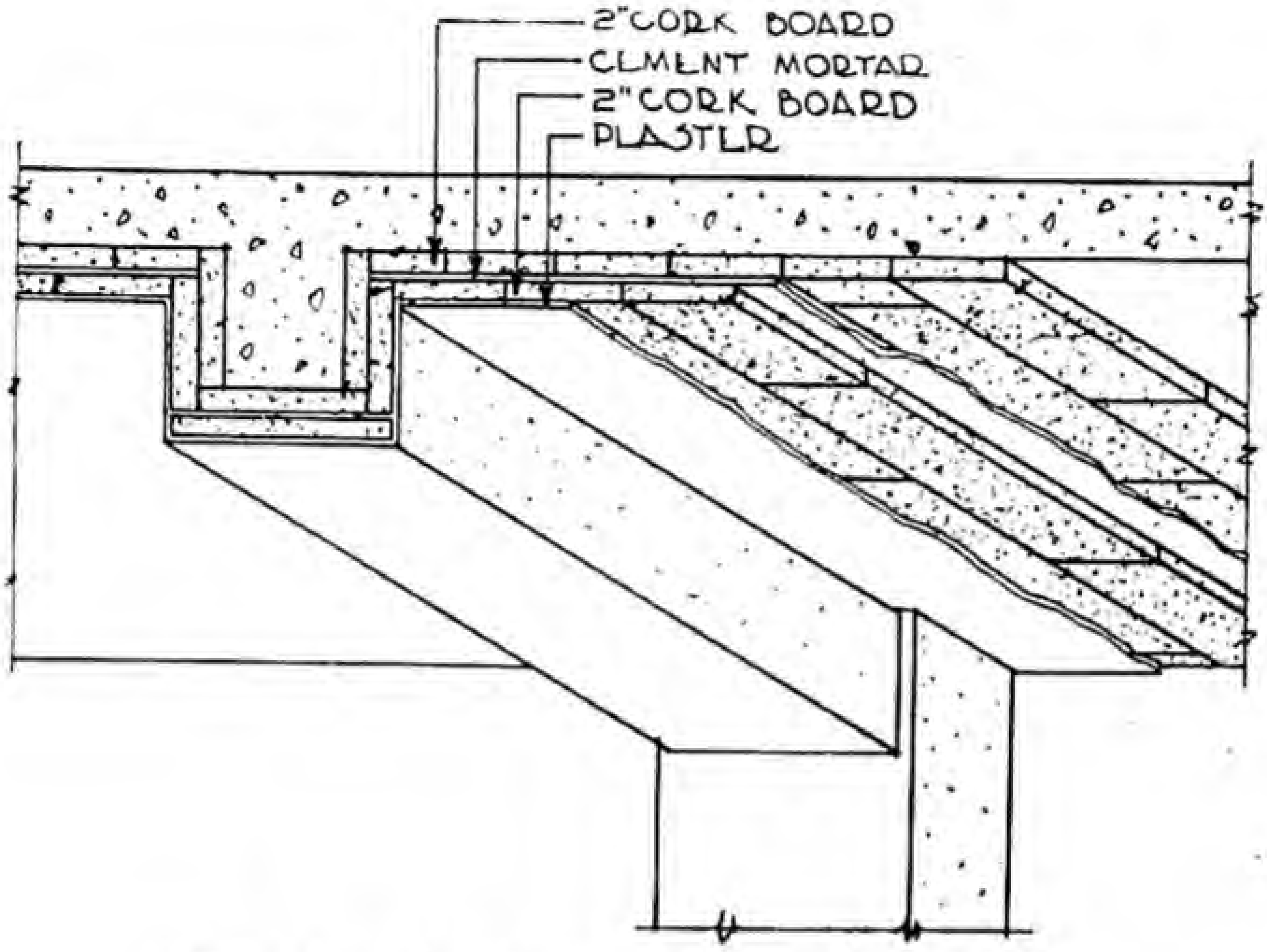


Fig. 166.—Insulation on Concrete Ceiling.

In the event that the top story of the building is refrigerated, the roof must be insulated. The roof may be insulated by putting the insulation on top of the roof slab, after which the roofing is applied directly upon the top of the insulation. Fig. 168 shows the detail of this construction.

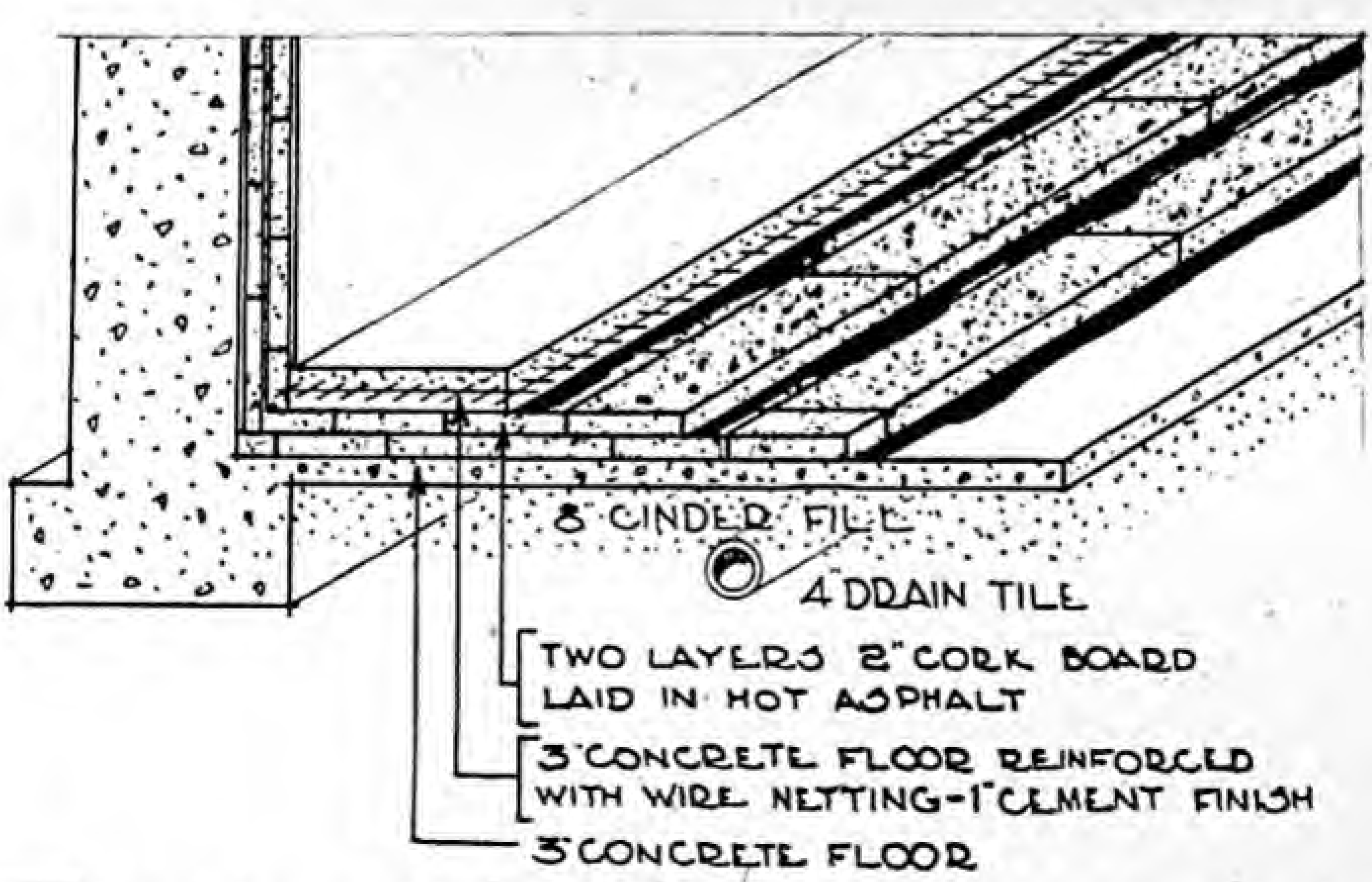


Fig. 167.—Diagram Showing Method of Insulating Ground Floor.

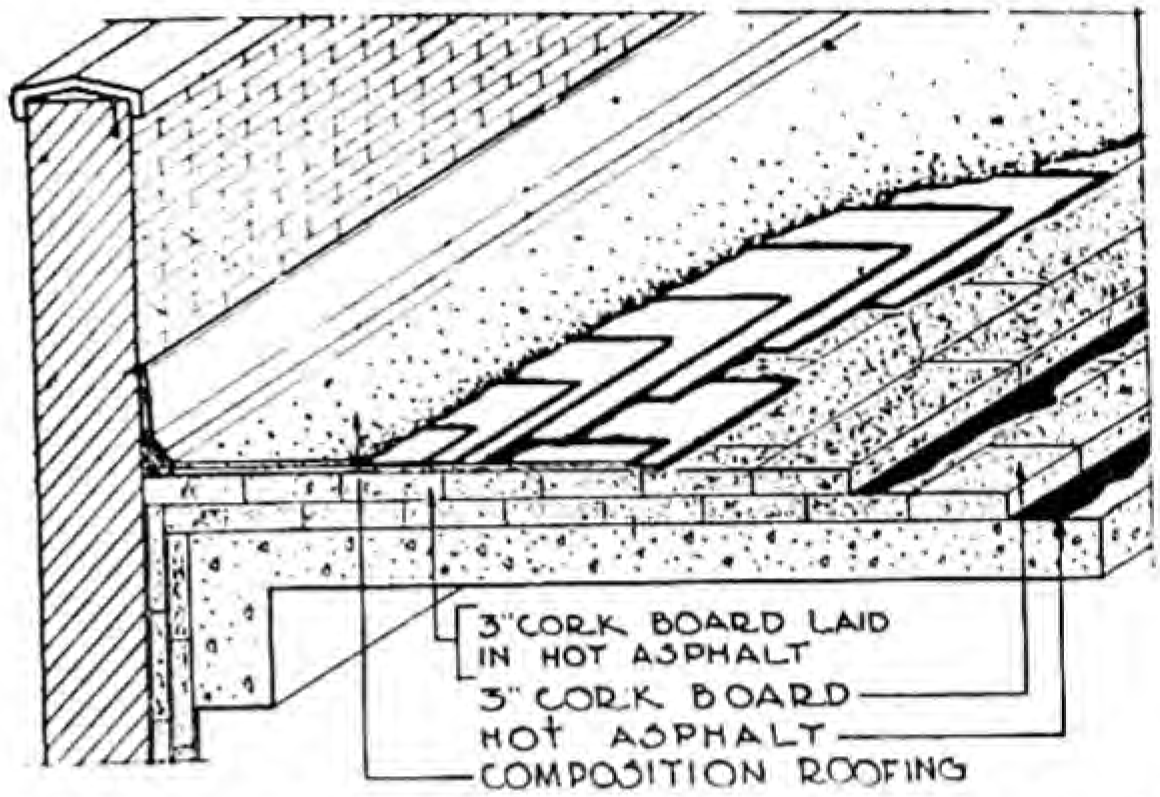


Fig. 168.—Diagram Showing Method of Applying Insulation to Roof.

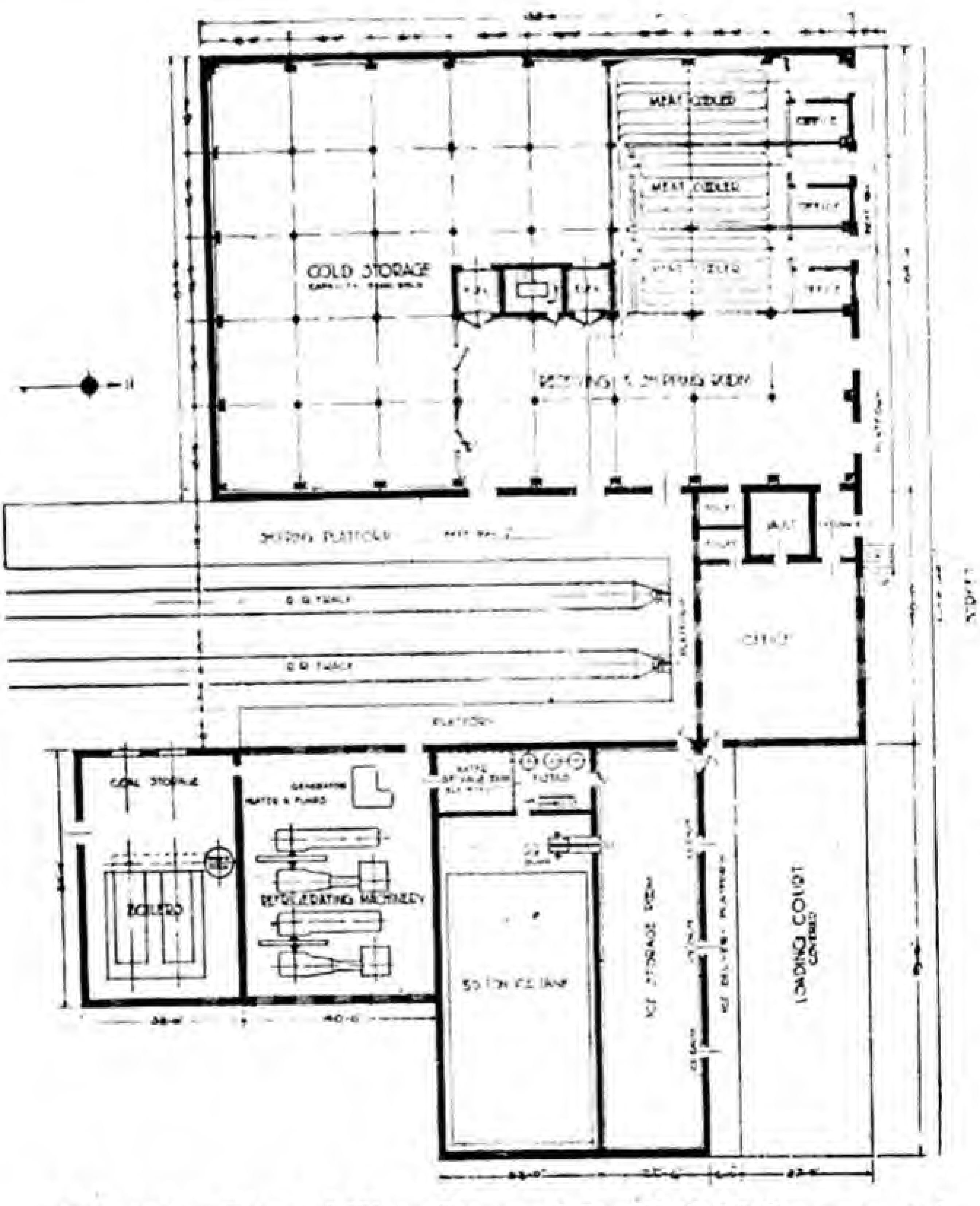


Fig. 169.—Plan of First Story of Small Cold Storage and Ice Making Plant.

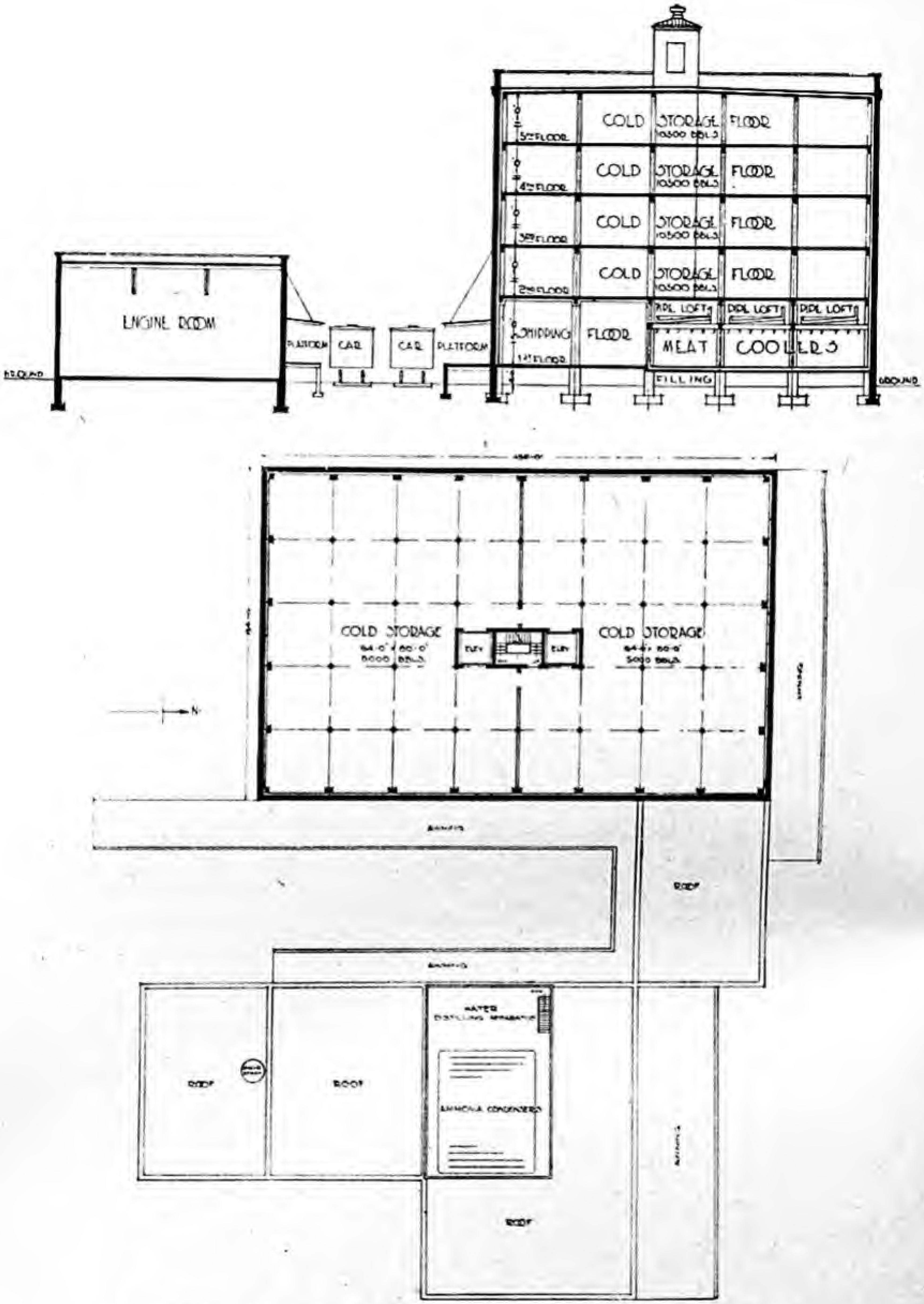


Fig. 170.—Plan and Elevation of Small Cold Storage and Ice Making Plant.

When rooms above each other are carried at different temperatures it is sometimes advisable to insulate the ceilings and the columns to prevent the transmission of heat, which would cause the formation of moisture upon the columns or the ceilings.

Typical Cold Storage Plants.—Figs. 162, 163 and 164 show the plans and elevations of the plant of the Merchants' Cold Storage & Warehouse Co., Chicago, Ill. Mr. H. P. Henschien was the architect employed to design the building, and the ammonia absorption refrigerating system was employed with the equipment on the top floor. A large number of the cold storage plants are not devoted exclusively to the cold storage of commodities, but rather in combination with the production of ice, etc. Also, it is evident that there are many storage plants in the United States which may be termed medium size. A small cold storage building, 195 ft. long, 84 ft. wide, and five stories high, in combination with a 50-ton ice making plant is shown by Figs. 169 and 170.

QUESTIONS ON CHAPTER XIV.

1. What are some of the general benefits of cold storage of food commodities?
2. What different temperatures are required in the different parts of cold storage warehouses and how do these temperatures affect the selection and type of refrigerating machine to be used for the production of refrigeration?
3. What are the most important food commodities which are placed in cold storage warehouses at present?
4. Name some of the important considerations in the selection of cold storage warehouse plants?
5. What types of building construction are most suitable for the construction of cold storage warehouses?
6. Describe the flat floor slab type of construction for floors of cold storage warehouses and enumerate its advantages.
7. Describe an efficient means of constructing the walls of cold storage warehouses.
8. How does the insulation affect the general design of the various building elements in the cold storage warehouse?
9. How should the insulation be incorporated into the floors, walls and ceilings?
10. What is the most economical and efficient method for the division of space in cold storage warehouses?

CHAPTER XV

COLD STORAGE OF COMMODITIES

Planning of Cold Storage Warehouses.—In the planning of cold storage warehouses it is evident that detailed consideration must be given to each viewpoint of the entire proposition. As indicated in the previous chapters, consideration must be given to the general location of the warehouse, giving especial attention to the service that may be rendered to the local merchants and to that which must be taken care of by the railroad. After the general location of the building has been determined, the next step in the planning of the project is the selection of the building type.

Also, as has been previously indicated, the type of insulation to be used and the proper method of incorporation into the building construction determine, to a large extent, the type of building construction that should be used. After the location and the type of building have been determined it is then necessary to give attention to the refrigeration required for the building, as well as efficient and economical means of producing refrigeration.

Refrigeration Duty.—As has been previously indicated, the most important items which combine to make up the total refrigerating load on a cold storage plant are as follows:

1. Refrigeration to cool the goods stored.
2. Refrigeration to absorb the heat transmitted through the insulation.
3. Refrigeration to offset the ventilation losses.
4. Refrigeration to absorb the heat generated in the room.

Goods may enter or leave cold storage in any of the following states: non-frozen, partly frozen, entirely frozen or sub-frozen, the latter indicating that the temperature is well below that of freezing. The amount of refrigeration required to handle the goods while in cold storage depends upon their weights, their specific and latent heats and the differences in their temperatures and states upon entering and leaving. The specific and latent heat of many materials commonly stored are given in Table 83 of this chapter. Unfortunately there are

many omissions in this table because the data is not available. It requires careful laboratory experiments to determine these values. It will be noticed, however, that the latent heat of fusion is proportional to the water content in a material and may be calculated approximately by multiplying the per cent of water by (144) the latent heat of water.

When a definite weight of any material is cooled from one temperature and state to another the heat removed during each process may be calculated as follows:

H_a = Btu. to cool above freezing from t_1 to $t_2 = W \times S_1 \times (t_1 - t_2)$
H_b = Btu. to cool above freezing from t_2 to $t_f = W \times S_1 \times (t_2 - t_f)$
H_c = Btu. to change state during freezing $= W \times X \times L$
H_d = Btu. to cool below freezing from t_f to $t_3 = W \times S_2 \times (t_3 - t_f)$

In the above formula:

W = Weight of the material in pounds.
S_1 = Specific heat before freezing in Btu. per lb.
t_1 = Initial temperature above freezing in ° F.
t_2 = Lower temperature above freezing in ° F.
t_f = The freezing temperature of the material in ° F.
X = Percentage of the material frozen.
S_2 = Specific heat after freezing in Btu. per lb.
t_3 = Final temperature below freezing in ° F.

The following example will illustrate the use of the above formulae:

Example: Find the Btu. removed from 10,000 lbs. of strawberries when cooling from 80° F. to 40° F. then from 40° F. to the freezing temp. 29° F. then to completely freeze them; then to cool them to —10° F. See Table 83 for specific and latent heats.

$$H_a = 10000 \times .92 \times (80\text{-}40) = 368{,}000 \text{ Btu.}$$
$$H_b = 10000 \times .92 \times (40\text{-}29) = 101{,}200$$
$$H_c = 10000 \times \frac{100}{100} \times 129 = 1{,}290{,}000$$
$$H_d = 10000 \times .47 \times (29\text{-}(\text{-}10)) = 183{,}300$$

Total 1,942,500 Btu.

Freezing temperatures of several varieties of fruits and vegetables are given in Tables 86, 87 and 88 at the end of this chapter. The presence of acids and salts in water tends to lower the freezing temperature of water, and food products containing a large percentage of water have freezing temperatures ranging from 25° to 31° F. An error resulting from using 29° F. in the above formula for freezing temperature will not seriously affect the results. Where the freezing temperature of a material is known it should be used in computations.

In the event that the amounts and kinds of material to be placed in cold storage rooms are not known, the heat required to cool the material must be estimated in some manner. This is generally done by estimating that so many Btu. per cu. ft. of room space must be removed in order to cool the materials which are stored in the storage room. The data in Table 84 gives values which may be used for this purpose.

TABLE 83.—COLD STORAGE DATA.

Name	Per Cent Water	Storage Temperature		Specific Heat		Latent Heat of Fusion
		Low °F	Normal °F	After Freezing	Before Freezing	
Ale	....	32	33	....	....	......
Apples	83	29	31	0.45	0.86	120
Apple Butter	....	....	40	....	....	......
Apricots	....	35	40	....	....	......
Asparagus	94	33	34	0.43	0.95	134
Bacon, Smoked Goods	....	30	32	....	....	......
Bananas	76	56	56	0.43	0.81	109
Beans, Green	89	32	33	0.47	0.92	128
Beans, Dried	12	32	40	0.24	0.30	18
Beef, Lean	72	30	32	0.41	0.77	102
Beef, Fat	51	30	32	0.34	0.60	72
Beef, Fresh, Chilling	68	30	32	0.40	0.74	98
Beef, Freezing	....	−30	0	....	....	......
Beef, Frozen	....	−10	0	....	....	......
Beef, Storage	15	....	33	0.27	0.34	22
Beef, Dried	....	....	40	....	....	......
Beer	....	32	33	....	....	......
Beer, Bottled	....	32	45	....	....	......
Berries	....	31	40	....	0.42	......
Butter, Tubs	13	0	15	0.24	0.30	19
Butter, Cartons	13	0	15	0.24	0.30	19
Buttermilk	....	....	40	....	....	......
Butterine	....	....	25	....	....	......
Buckwheat Flour	....	....	42	....	....	......
Bulbs	....	23	35	....	....	......
Cabbage, Bulk	91	25	31	0.43	0.93	129
Cabbage, Boxed	....	25	31	....	....	......
Canned Goods	....	....	35	....	....	......
Cantaloupes	45	33	36	0.34	0.56	65
Carrots	83	30	36	0.45	0.86	120
Cauliflower	93	32	....	0.48	0.94	134
Celery	95	10	33	0.49	0.96	137
Cereal Foods	....	40	45	....	....	......
Cheese, Cream	35	30	32	0.30	0.48	50
Cheese, Brick	35	30	32	0.30	0.48	50
Cherries, Dried	....	....	45	....	....	......
Cherries, Fresh	82	....	40	0.45	0.86	118
Chocolate, Dipping	....	....	65	....	....	......
Chocolate, Storage	....	33	42	....	....	......
Cider	....	30	32	....	....	......
Cigars	....	....	35	....	....	......
Codfish, Dried	....	....	40	....	....	......
Condensed Cream	....	33	38	....	....	......
Corn, Dried	11	35	45	0.23	0.28	15
Corn, Green	75	....	38	0.43	0.80	108
Cranberries	....	28	33	....	....	......
Cream, Fresh	59	32	34	0.38	0.90	84
Cucumbers	95	32	38	....	....	......
Currants, Dried	....	....	45	....	....	......
Currants, Fresh	....	....	38	....	....	......
Chestnuts	....	....	34	....	....	......
Dates, Cured	79	....	40	0.44	0.83	104
Eggs, Freezing	70	−10	0	0.40	0.76	100

TABLE 83.—COLD STORAGE DATA—(Continued)

Name	Per Cent Water	Storage Temperature		Specific Heat		Latent Heat of Fusion
		Low °F	Normal °F	After Freezing	Before Freezing	
Eggs, Storage	70	28	29	0.40	0.76	100
Fruits, Dried	30	35	40	0.32	0.47	43
Fruits, Canned	....	40	....	....	....	......
Figs, Dried	....	45	....	....	....	......
Filberts	....	40	....	....	....	......
Fish, Dried	45	35	36	0.34	0.56	65
Fish, Freezing	73	—15	....	0.43	0.82	111
Fish, Storage	....	....	26	....	....	......
Fish, Frozen	....	0	10	....	....	......
Flour and Meal	20	36	40	0.28	0.38	29
Furs and Fabrics	....	25	35	....	....	......
Furs, Undressed	....	....	35	....	....	......
Flowers, Cut	....	....	36	....	....	......
Game, Freezing	60	5	10	0.38	0.68	87
Game, Storage	....	15	25	....	....	......
Grapes, Dried	....	30	40	....	....	......
Grapes, Fresh	58	26	32	0.38	0.67	84
Grapefruit	....	32	36	....	....	......
Ice Cream, Freezing	67	0	15	0.40	0.74	96
Ice Cream	67	5	15	0.40	0.74	96
Ice	....	20	28	0.508	1.00	144
Lard	....	32	38	0.31	0.54	......
Lemons	63	36	38	0.39	0.70	90
Lettuce	94	26	42	0.48	0.95	135
Lily of Valley Pips	....	....	24	....	....	......
Lobsters	77	....	25	0.42	0.81	108
Melons	....	33	40	....	....	......
Milk, Fresh	88	32	36	0.47	0.90	124
Molasses	....	42	45	....	....	......
Mutton, Chilling	....	30	32	0.67	0.81	100
Meats, Canned	....	....	40	....	....	......
Nuts	10	....	35	0.24	0.28	14
Nursery Stock	....	28	30	....	....	......
Oleomargarine	....	20	25	....	....	......
Onions	88	32	35	0.46	0.90	127
Oranges	63	32	35	0.39	0.70	90
Oysters, Shell	80	30	35	0.44	0.84	114
Oysters, Bulk, Iced	87	20	25	0.46	0.90	125
Parsnips	83	32	33	0.45	0.86	120
Peaches, Dried	....	....	40	....	....	......
Peaches, Fresh	87	30	30	0.46	0.90	125
Pears, Dried	....	....	40	....	....	......
Pears, Fresh	83	30	32	0.45	0.86	120
Peas, Dried	10	35	40	0.22	0.28	14
Peas, Fresh	75	32	36	0.42	0.80	108
Pineapple	....	32	40	....	....	......
Plants, Potted	....	30	35	....	....	......
Plums, Dried	....	....	40	....	....	......
Plums, Fresh	84	28	32	0.45	0.87	121
Pork, Salt	....	....	42	....	....	......
Pork, Chilling, Fat	39	30	32	0.30	0.51	55
Pork, Hams	60	29	32	0.38	0.68	87
Pork, Freeze	....	—20	0	....	....	......

TABLE 83.—COLD STORAGE DATA—(Concluded)

Name	Per Cent Water	Storage Temperature Low °F	Storage Temperature Normal °F	Specific Heat After Freezing	Specific Heat Before Freezing	Latent Heat of Fusion
Pork, Freezer, Storage	....	0	10	....	....	......
Potatoes	73	30	33	0.42	0.78	105
Potatoes, Sweet	69	50	55	0.41	0.75	99
Poultry, Freeze	74	0	10	0.42	0.80	105
Poultry, Storage	....	28	30	....	....	......
Poultry, Frozen	....	−10	0	....	0.377	......
Pigeons	72	28	30	0.41	0.78	102
Radishes	92	28	33	0.48	0.94	132
Rice	....	20	42	....	....	......
Sauerkraut	....	35	36	0.47	0.92	129
Strawberries	90	33	40	0.47	0.92	130
Tomatoes	94	33	34	0.48	0.95	135
Vegetables	....	30	33	....	....	......
Veal	63	....	34	0.39	0.70	90
Watermelons	....	33	36	....	....	......
Wines	....	32	45	....	....	......
Yeast	....	28	....	....	....	......
Woolens	....	15	28	....	....	......

The losses of refrigeration through the cold storage room walls depend upon the kind of insulation, the thickness of the insulation, and the relative temperature difference between the inside and the outside of the room. The thickness of the insulating material and the relative kind of material determine the rate of heat transfer per unit of area and time. As has been previously indicated in Chapter VIII pertaining to heat transmission in insulation and apparatus, the heat that is transmitted through a given wall of insulation depends primarily upon the internal resistance due to the conductivity of the material.

It was also emphasized that the resistance due to the flow of heat at the surface of the materials would amount to five to ten per cent of the total resistance. Since the values for the internal thermal conductivity of insulating material have been determined under laboratory

TABLE 84.—HEAT TO COOL MATERIALS STORED IN SMALL REFRIGERATORS AND COLD STORAGE ROOMS.

Storage room temperature	B.t.u. per hour per cu. ft. of room space
5-10	5.0
10-15	4.0
15-20	3.5
20-25	3.0
26-30	2.5
31-35	2.0
36-40	1.5
41-50	1.25
51-65	1.10

conditions which are most favorable to the efficiency of the insulation, and since in the actual cold storage plant the insulating materials may be subjected to adverse conditions, it is usually advisable to apply a certain factor of safety after the theoretical amount of heat flow has been determined.

In the event that the heat transmission coefficient has been estimated by taking into consideration the resistance to the flow of the heat at the surface of the materials, it is advisable to increase this coefficient by 25 to 35 per cent; in the event that the coefficient of heat transmission has been taken to be proportional to the internal conductivity alone, the coefficient thus determined should be increased from 15 to 25 per cent, depending in both cases upon the conditions in the plant. In a plant in which the walls are constructed in an efficient manner and where provision has been made for thoroughly waterproofing the material, the lowest factors of safety may be used; but, on the other hand, when the wall is not so well constructed or waterproofed, the highest factors of safety should be used.

The foregoing facts should be taken into consideration when determining the heat transfer coefficient, K, for any cold storage wall. After the magnitude of the heat transfer coefficient has been determined the total heat transfer for a given wall with a given temperature difference may be obtained as follows:

$$H = K \times A \times (t_1 - t_2)$$

in which H = heat transmitted by the insulation
K = heat transfer coefficient in Btu. per hr. per sq. ft. per deg. of temp. diff., as determined by methods indicated in Chapter VIII.
A = area of wall in sq. ft.
t_1 = outside air atmospheric temperature
t_2 = temperature in cold storage room

The resistance of the main cold storage wall to the flow of heat is seldom taken into consideration, except in the more refined calculations. Ordinarily, the heat transfer coefficient, K, will be determined by the thickness and kind of insulation only.

The heat to be taken from the air which is admitted for ventilation purposes depends upon the size of the room, the temperatures of the room and the outside air, and the relative humidity of the air. The cooling required amounts to the sensible heat required for cooling the air, and the latent heat of condensation of the moisture that is condensed out of the air when the air is cooled.

The amount of air required for ventilation, of course, will vary with the nature of the products stored. In the ordinary cold storage room the volume of air that is necessary for ventilation purposes is not a well defined factor at present. The amount of air required will depend upon the production of foul gases and odors in the room. In

ordinary rooms in which such products as chilled meats, poultry, game, fruits, vegetables, eggs, dairy products, etc., are stored the air admitted to the room for ventilation purposes and that due to opening of doors and leakage through the windows may be taken from Table 85. This table gives the number of air changes per hour for rooms of various sizes:

TABLE 85.—AIR FOR VENTILATION IN COLD STORAGE ROOMS.

Size of room in cu. ft.	No. of air changes per hour
500- 900	1.0
1,000- 1,500	0.5
1,600- 2,000	0.4
2,100- 2,900	0.3
3,000- 5,000	0.2
5,500- 10,000	0.1
10,500- 20,000	0.05
20,500- 40,000	0.025
40,500-100,000	0.01

After the volume of air admitted to the cold storage room has been ascertained the weight of the air in pounds may be determined by dividing the volume in cubic feet by the specific volume of one pound of air. This may be expressed symbolically as follows:

$$W = \frac{V}{v_s}$$

in which W = weight of air in pounds
V = volume of air in cu. ft.
v_s = specific volume of air in cu. ft. per lb.

The specific volume of air for various temperatures is given by Table 90, Chapter XVI. Column three of this table gives the volume of one pound of air saturated with water vapor. This is accurate enough for practical purposes. The specific volume should be taken for the air at the temperature at which it enters the cold storage room, which is in general the temperature of the atmosphere. After the weight of the air for ventilation purposes has been determined, the heat required to cool this amount of air will be the product of the weight of the air, the specific heat, and the temperature range. However, this calculation only gives the heat required to cool the air from the higher temperature to the lower temperature, as sensible heat, and does not take into consideration the latent heat of condensation of the moisture which will be condensed out of the air as it is cooled.

The amount of heat required for condensing the moisture will depend upon the relative humidity of the atmospheric air, and the temperature to which the air is cooled. The heat required to cool air saturated with moisture from a higher temperature to a lower temperature

may be taken from Table 90, Chapter XVI. The last column in this table gives the heat content of a pound of air saturated with water vapor at the various temperatures. By noting the difference in heat content between the two temperatures, the heat equivalent to the sensible heat of the air as well as the heat required for the condensation of the moisture will be determined. This may be expressed in a formula as follows:

$$H = W \times (h_1 - h_2)$$

in which H = total heat required to cool the air
W = weight of air in lbs.
h_1 = heat content at higher temperature
h_2 = heat content at lower temperature

It will be noted that Table 90 of Chapter XVI gives the heat content for air that is saturated with moisture. This is not quite the condition of the air as it is admitted to the coldstorage room. However, during the summer time, the relative humidity may be as high as 70 to 90 per cent, so that when the refrigeration required is calculated as above outlined, a little more than the actual amount of heat required will be found. This is due to the fact that as soon as air that is saturated is cooled, the moisture begins to condense out, but when we have atmospheric air being cooled, which has a humidity of 70 to 90 per cent, it must be cooled a few degrees before the moisture begins to condense. It should be remembered, however, that the air required for ventilation purposes in the rooms will vary with the nature of the refrigeration work being produced. For example, in packing houses the meat chilling rooms will require from two to six changes of air in the room per twenty-four hours.

From the foregoing, it will be observed that the amount of ventilation required in general cold storage rooms will vary with the nature of the goods stored, operating conditions, etc. The air that is admitted for ventilation brings in moisture with it from the outside atmosphere and this moisture, upon being cooled, is condensed on the refrigerating coil surfaces. Much of the odors and foul gases that are given off in cold storage rooms are absorbed by this moisture and thus removed from the air in the room.

For assisting the purification of the air in the room, a few windows should be installed which may be opened for the purpose of ventilation. Of course, the doors and windows should be constructed in a workmanlike manner so that there is very little air leakage through these during the hot summer months.

In some cases, which is especially true of the larger rooms, it is desirable to use the forced air circulation for the distribution of refrigeration. This system consists of a refrigerating coil located in a bunker, a fan which is generally direct connected to an electric motor,

and an air duct system. The fan is generally arranged so that the air is drawn in from one-half of the duct system through the room containing the refrigerating coils to the fan. The fan takes in the cooled air and discharges it into the discharge air duct system. The air duct system should be arranged so that there is a vigorous circulation of air in all parts of the room. In the use of the forced air circulation system, the air that is cooled may be taken directly from the room, or all or part may be taken from the outside atmosphere. In general, such a circulating system takes the air from the cold storage room, cools it a few degrees, and discharges the air back into the cold storage room. The volume of air, of course, depends directly upon the conditions in the room, the magnitude of the refrigeration load, the moisture conditions, etc.

Further consideration regarding the cooling of air in cold storage rooms will be given later in the chapter pertaining to the cooling of air. At present, the forced air circulating system for cold storage rooms is being installed in some of the new plants in the United States. It seems that the gravity air circulating system is also used extensively in both the large and small rooms. It is evident that both systems have their advantages and disadvantages.

In estimating the refrigeration requirements for cold storage rooms, allowance must be made for any heat that is generated within the room. Heat may be generated by workmen, lights, motors, fans or any other heating device. The heat given off by men working in cold storage rooms will vary from 400 to 600 Btu. per hour, depending upon the nature of the work being performed. An average of 500 Btu. per hr. may be used for this purpose.

The heat given off by electric lights may be determined by remembering that one horsepower is equal to 2545 Btu. per hr., and that this is equivalent to 746 watts. The heat per watt of capacity in the electric lights per hour would then be determined by dividing 2545 by 746, which equals 3.41. The following tabulation shows the heat in Btu. given off per hr. for various capacity electric lights:

Watts capacity in electric lights	Btu. per hr. per electric light
25	85.3
50	170.5
100	341
200	682
400	1364
600	2046

An estimate of the amount of lights required in a cold storage room may be made by allowing 0.5 to 1.0 watt per sq. ft. of floor area. The heat equivalent of any work performed in a cold storage room by motors, fans or any other power dissipating apparatus may be estimated

by remembering that one horsepower is equivalent to 2545 Btu. per hr. which, in turn, is equal to 42.42 Btu. per min. Any other-heat that is allowed to get into the storage room should be allowed for in direct ratio to the magnitude of the heat, and it is evident that all unnecessary heat should be prevented from entering the room, since this heat must be removed by the refrigerating system.

Cold Storage Data.—At present, there seems to be considerable lack of knowledge pertaining to data on miscellaneous considerations regarding the cold storage of various products. This refers in particular to the specific heats, the latent heats of fusion, specific gravities, units of storage, weights per unit, weight per cubic foot of storage space, weight per square foot of storage space, space occupied per unit, etc. The following considerations pertain to the space occupied by various units of common cold storage commodities, the weight of same, etc.:

A barrel may be taken to contain 31.5 gals., while the U. S. standard gallon contains 231 cu. in. A bushel may be taken as being equivalent to 1.245 cu. ft.

Apples may be stored in boxes which are approximately 10½ in. x 11½ in. x 18 in., which is commonly termed a bushel box.

Apples are also stored in barrels which contain 2½ bushels, and weigh approximately 150 lbs. per barrel. These barrels occupy approximately 5 cu. ft. of space.

The desirable storage temperature for apples varies from 29° to 31° F. Apples seem to improve when placed in cold storage for a comparatively short time, due to the fact that some of the starch is transformed into sugar.

Butter is usually stored in tubs which contain from 50 to 60 lbs., and which occupy about 2 cu. ft. of space. Butter may be held in cold storage for a short time at a temperature of 32° to 33° F. If it is carried for a comparatively long time it should be frozen and then carried in a freezer storage at 15° F. or lower. The flavor of butter is not improved by cold storage, but will slightly deteriorate after considerable time. Since butter will absorb odors, it is evident that the containers as well as the rooms must be entirely clean and free from odors. Butter in tubs should be protected from direct contact with the air. It is generally not advisable to store the butter in cartons. The humidity may be carried at a fairly high point, and a vigorous circulation of the air about the room should be maintained to carry the odors and other gases to the freezing surface of the refrigerating coils. The air in the room may be further purified by periodical ventilation of the room. The various parts of the room should be covered with a suitable white wash.

Celery is usually stored in crates which contain about 140 lbs. and which have dimensions of about 24x24x30 in. They occupy about 10 cu. ft. per crate. Celery is generally held at a temperature of 33° to 34° F.

Cheese is usually stored in boxes which contain about 60 lbs. and will occupy about 2 cu. ft. Desirable storage temperatures for cheese seems to vary between 30° and 32° F. Refrigeration is also used in the ripening process in the manufacture of cheese. Storage after the cheese has been ripened does not improve its quality.

Eggs are usually stored in cases which contain about thirty doz. and which weigh 50 to 55 lbs. The weight of a dozen eggs may be taken as 1¼ lbs., and when eggs are broken, about 90 eggs will make a gallon. The crates have dimensions of 12x13x25 in. and occupy about 2¼ cu. ft. of space. Eggs lose weight upon being held in cold storage for a few months, amounting to about 7 per cent for the first six months. The loss in weight may be reduced by regulating the humidity. The desirable temperature for the storage of eggs varies from 28° to 29° F. Eggs may have the shells removed and put into suitable containers, after which they are frozen. Desirable freezing temperatures for eggs will vary from 0° to −20° F. After they have been frozen, they may be stored in a room at −10° to 0° F. When they are removed from the freezer storage they should be allowed to thaw slowly at a temperature of 40° to 45° F. Also, eggs absorb odors, therefore they should be placed in rooms which contain eggs only.

Eggs, upon being placed in a cold storage room, should be cooled slowly, after which they should be held at a uniform cold storage temperature. The desirable humidity for eggs which are stored at a temperature of 28° to 30° F. seems to vary from 80 to 85 per cent. The egg storage room should be ventilated frequently when the room is being filled. After the eggs have been cooled to the cold storage room temperature, only periodical ventilation is necessary. Rooms in which the eggs are candled or broken should be maintained at temperatures varying from 55° to 60° F. for best results.

Fish is usually frozen at a temperature of approximately −15° F. This is accomplished by placing the fish in pans which are placed directly upon the refrigerating piping, which acts as shelves. The size of the pans are generally about 22x8x2½ in. deep, and hold about 12 lbs. After they have been frozen solid they are coated with a thin layer of ice by dipping them in water at a low temperature. After this operation, they are prepared for shipment in a room which is maintained at a temperature of 10° to 15° F.

Nearly all kinds of fruits may be held in cold storage for short periods of time. In general, the storage time should not be long since the majority of fruits will lose flavor in long storage.

Meats are generally improved by storing in cold storage for comparatively short periods, ranging from one to three weeks, after which the meat will gradually lose its flavor. The desirable storage temperature seems to vary from 28° to 32° F., depending upon the kind of meat stored. The weights of the various animals may be taken from the following tabulation:

Beef	750 lbs.
Calves	90 lbs.
Sheep	75 lbs.
Hogs	250 lbs.

Meat is very often held in cold storage for long periods of time by freezing it. The temperatures of freezing used for meats will vary from 5° to 10° F. on down. The meat is arranged in the cold storage room so that the air may circulate freely about all of the parts.

Poultry is generally held in storage at a temperature of 28° to 30° F. The average weight of chickens may be taken as 5 lbs. If it is desired to hold the poultry in cold storage a considerable length of time, it should be frozen at a temperature of 0° to 10° F., after which it should be stored in a room having a temperature of from 0° to 10° F. Poultry is usually placed in small boxes or barrels.

Oranges and lemons are generally stored in a room which has a temperature of 32° to 35° F. Oranges are packed in crates which have dimensions of 15x15x30 in. and which weigh about 70 lbs.

Potatoes are stored in rooms which have a temperature varying from 30° to 33° F. They are usually stored in barrels or sacks. The barrels generally contain about 2½ bushels and occupy space of about 5 cu. ft. Potatoes weigh about 60 lbs. per bushel, or about 180 lbs. per barrel, which includes the weight of the barrel.

In determining the size of the room to be used in storing a certain commodity, the method of placing the material in the room, the nature of the package, etc., must be taken into consideration. The nature of the package of the material determines generally the height to which the packages may be piled. Generally, the small packages of materials will be placed in piles which do not exceed the height of 5 to 6 ft. Barrels and the larger packages may be piled together until the total height of 8 to 10 ft. is reached. In determining the amount of space which is necessary, about one-third to one-fifth should be allowed for aisles, piping, etc. In the selection of materials for storage at low temperatues, it is obvious that only materials of first quality should be selected for storage. The quality of the goods stored, together with the temperatures of the room and the other conditions, determine whether or not the storage of such materials will be successful.

Table 86 gives the freezing points for various kinds of fruits. The

freezing points of potatoes, sweet potatoes, tomatoes and other vegetables are given in Table 88.

Refrigerating Equipment for Cold Storage.—The refrigerating equipment that is installed in the medium and large sized cold storage plants consists usually of the ammonia compression or absorption refrigerating system. In small cold storage warehouses in which the temperatures are fairly high the simple ammonia compression system may be used. However, in the majority of the cold storage warehouses

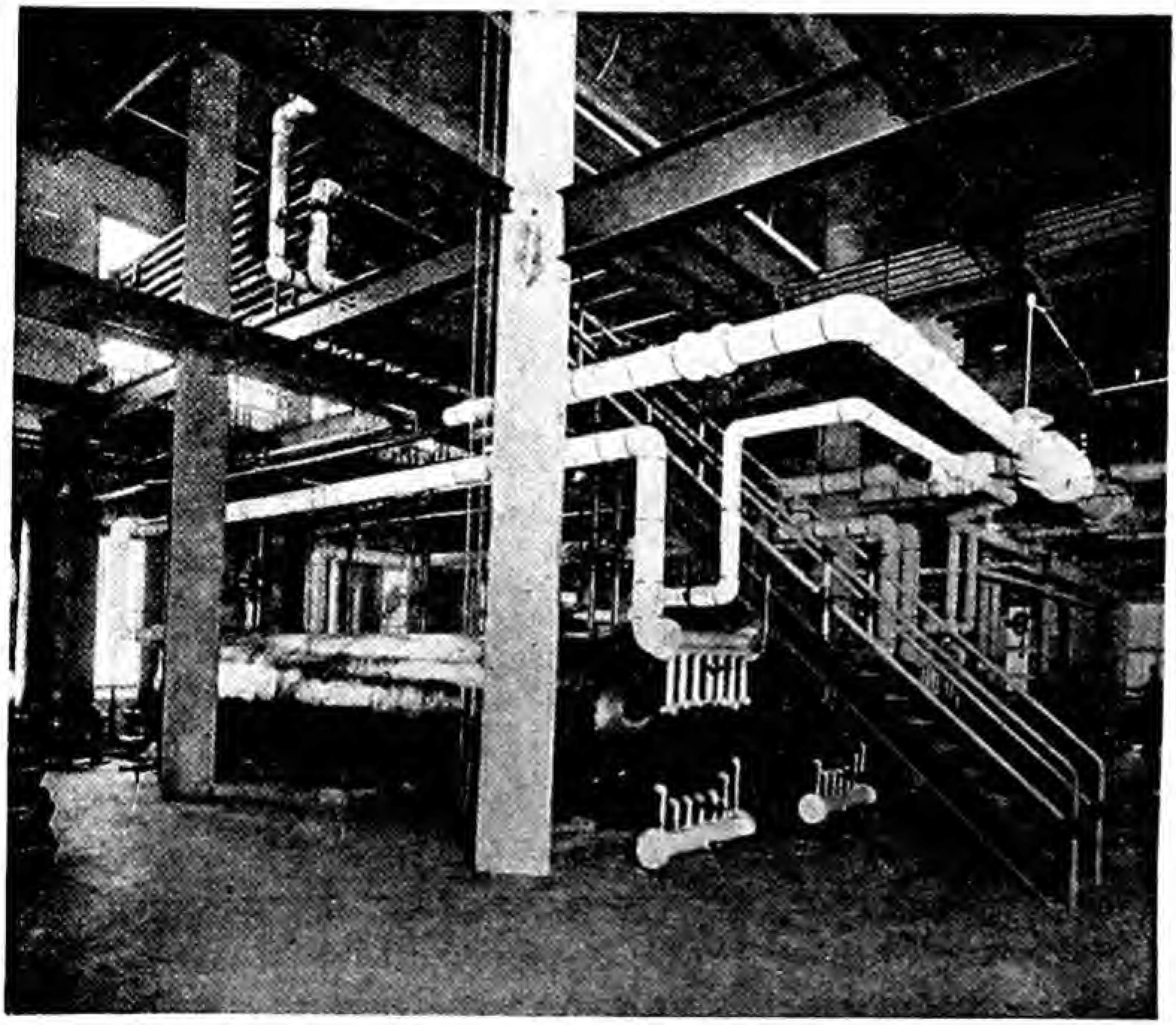


Fig. 171.—Absorption Machine in Cold Storage Plant.

of average size temperatures varying from $-15°$ F. to $35°$ F. must be maintained. For the production of these lower temperatures it is usually advantageous to use the absorption refrigerating machine or the two-stage ammonia compression refrigerating system. In order to produce these lower temperatures the suction pressure on the machine must be quite low to secure a very low boiling temperature of the ammonia. If the simple ammonia compression system is used for these lower suction pressures, the plant will be found to be expensive to

operate, because the horsepower required increases rapidly as the suction pressure is lowered. This accounts for the extensive use of the ammonia absorption or the two-stage ammonia compression refrigerating systems for such work.

The plant of the Merchants' Cold Storage & Warehouse Co., previously mentioned in this chapter, is refrigerated by means of the ammonia absorption refrigerating system. The refrigerating machinery is located on the top floor, as shown in Fig. 164. Three units, each having a capacity of 150 tons of refrigeration per day, were installed. These units are operated by means of exhaust steam. Fig. 171 shows a view of the machine room on the top floor of the building. The ammonia generators are of the horizontal type with horizontal heating coils arranged for the exhaust steam. The absorbers are of the horizontal tubular type and are arranged for twenty passes of cooling water. The exchangers are of the vertical shell and coil type, each 150-ton unit being supplied with two 75-ton exchangers. The aqua ammonia pumps are the single direct double-acting steam-driven type, which are controlled automatically by the liquid level in the absorber by means of suitable float valves.

The storage rooms are refrigerated by means of the circulation of brine which is cooled in three horizontal tubular brine coolers, 5 ft. in diameter and 18 ft. long. Calcium chloride brine is used and may be cooled to the temperatures of $-20°$ F., $0°$ F., and $15°$ F. in the various coolers. These coolers are arranged for ten passes of brine. The brine is circulated throughout the storage room by means of three 8-in. pumps, the 3-pipe balanced system of connections being used. The brine pumps take the brine from the balance tanks which are located above the highest point of the system, and discharge the brine through the brine coolers and thence to the refrigerating coils in the various storage rooms, after which it is collected by the piping sytem and returned to the balance tanks.

The remainder of the refrigerating equipment is located on a platform just above the generators, absorbers and brine coolers. The rectifiers are of the vertical double-pipe type, consisting of 2- and 4-in. pipe. The condensers are of the double-pipe type and are constructed of 1½- and 2½-in. pipe. The weak liquor coolers are of the double-pipe type, constructed of 2- and 3-in. pipe. In addition to cooling the storage rooms, the equipment is arranged for producing ice in a tank which is located in a nearby building. The various units are cross-connected to secure flexibility of operation. The refrigeration equipment required approximately 35 lbs. of steam per hr. per ton of refrigerating effect, and used two gals. of water at $60°$ F.

TABLE 86.—AVERAGE AND EXTREME FREEZING POINTS OF FRUITS.

R. C. WRIGHT and G. F. TAYLOR, U. S. Dept. Agriculture

Fruit and varieties	Temperatures (° F.) Average	Extremes Minimum	Extremes Maximum
Apples, summer varieties:			
Yellow Transparent	27.72	27.29	28.16
Red Astrachan	28.58	28.25	28.70
Early Ripe	29.18	28.82	29.47
Red June	29.59	29.29	29.71
Sweitzer	27.38	27.32	27.41
Shoemaker	28.46	27.93	28.03
Benoni	28.83	28.49	29.00
Early Joe	27.81	27.60	28.49
Martha (crab)	26.70	26.62	26.76
Average (not including the crab apple)	28.44	28.12	28.62
Apples, fall and winter varieties, eastern grown:			
Baldwin	29.04	28.84	29.43
Ben Davis	28.61	28.21	28.96
Delicious	28.48	28.16	29.10
Grimes	28.97	28.82	29.05
Jonathan	28.22	27.79	28.69
Paragon	28.50	28.45	28.55
Rambo	28.55	28.34	28.90
Stayman Winesap	28.51	28.02	28.91
Winesap	28.23	27.93	28.72
Yellow Newtown	28.00	27.80	28.20
York Imperial	28.34	28.10	28.50
Average	28.49	28.22	28.82
Apples, fall and winter varieties, western grown:			
Delicious	28.36	27.98	28.86
Gano	28.55	28.26	29.05
Grimes	28.60	28.26	29.05
Jonathan	28.35	28.02	28.72
Rome Beauty	28.92	28.72	29.38
Esopus (Spitzenberg)	28.69	28.26	29.05
Winesap	28.24	27.93	28.35
Average	28.53	28.20	28.92
Cherries:			
Early Richmond	27.94	27.60	28.35
Montmorency	28.10	27.79	28.58
St. Medard	28.09	27.60	28.58
Royal Nouville	28.16	27.95	28.50
Gloire de France	27.65	27.37	28.21
Mecker	26.88	26.76	27.69
Bigarreau (unknown variety)	27.83	27.83	27.83
Average	27.81	27.56	28.25
Grapes:			
American varieties—			
New Concord	28.39	27.93	28.68
Ambrosia	28.21	27.83	28.63
Dracut Amber	27.88	27.77	28.10
Moores Early	28.28	28.15	28.62
Captivator	27.86	27.14	28.05
Campbell (black)	27.94	27.77	28.00
Mericadel	28.54	28.40	28.54
Average	28.16	27.85	28.37
European varieties—			
Malaga	24.60	24.60	24.80
Emperor	24.60	24.10	24.76
Average	24.60	24.35	24.78
Oranges:			
Temple	28.64	28.34	28.82
Pineapple	27.72	27.60	27.83
Florida Seedling	28.20	28.10	28.43
Washington Navel	28.42	28.30	28.68
Oranges—Continued.			
Valencia (California)	27.01	26.90	27.60
Satsuma (Owari variety)	28.18	27.93	28.68
Average	28.03	27.86	28.34
Peaches (hard ripe):			
Belle	29.82	29.50	30.28
Elberta	29.72	29.43	30.00
Stevens	28.65	28.25	28.90
Edgemont	29.40	29.30	29.50
Williams	29.56	29.10	30.00
Bilyeu	28.90	28.35	28.98
Smock	29.28	29.05	29.57
Salwey	29.57	29.10	29.80
Hiley	30.02	29.90	30.24
Carman	29.57	29.30	29.95
Champion	29.06	28.73	29.95
Average	29.41	29.09	29.74
Plums:			
Burbank	29.26	29.05	29.80
Wickson	29.53	29.19	29.75
Tragedy	27.21	26.76	27.41
Red June	28.13	27.79	28.44
Average	28.53	28.20	28.85
Strawberries:			
American	29.70	29.66	29.75
Big Late	30.03	29.25	30.05
Big Joe	29.98	29.78	30.19
Brandywine	29.96	29.85	30.36
Chesapeake	30.29	29.94	30.32
Dunlap	29.82	29.24	29.99
Excelsior	29.94	29.28	30.04
Early Ozark	29.82	29.66	30.13
Early Jersey Giant	29.82	29.43	30.22
Gandy	29.24	28.85	29.55
Glen Mary	30.08	29.53	30.16
Howard 17 (Premier)	30.23	29.58	30.38
Hustler	30.48	30.41	30.60
Klondike	29.59	29.28	29.90
Kellog (Kellog's Pride)	30.13	29.78	30.48
Late Jersey Giant	30.25	30.13	30.28
Lupton	28.84	28.82	29.10
Rewastico	30.05	30.03	30.13
Stevens	30.18	29.37	30.42
Sample	30.38	29.63	30.48
Superb	30.46	29.85	30.81
Twilley	29.22	28.98	29.53
Average	29.93	29.56	30.13
Blackberries:			
Jumbo	29.09	28.71	29.30
Eldorado	29.21	28.76	29.54
Crystal White	28.40	28.12	28.63
Logan (Loganberry)	29.51	29.32	29.75
Raspberries:			
Ranere (St. Regis, red)	30.41	30.12	30.50
Columbia (black)	28.76	28.24	28.79
Cranberries:			
Searl	28.20	27.93	28.44
Gebhart Beauty	26.30	26.00	26.60
Mammoth	26.70	26.40	26.90
Metallic	25.60	24.80	25.80
Chipman	26.89	26.01	27.36
Perry Red	27.93	26.62	28.05
Early Black	28.10	27.64	28.71
McFarlin	29.02	28.38	29.45
Shaw's Success	25.03	24.62	25.74
Howes	28.24	27.50	28.43
Pride	27.05	26.57	27.73
Wales Henry	28.70	27.92	28.00

TABLE 87.—SUMMARY OF AVERAGES.

Apples:				Grapefruit	28.36	28.00	28.50
Summer varieties	28.44	28.12	28.62	Lemons	28.14	27.89	28.47
Fall and winter	28.51	28.21	28.87	Oranges	28.03	27.86	28.34
Bananas (Jamaica):				Peaches (hard ripe)	29.41	29.09	29.74
Green—Peel	29.84	29.76	29.92	Pears (Bartlett):			
Green—Pulp	30.22	30.10	30.58	Hard ripe	28.46	28.06	28.70
Ripe—Peel	29.36	29.15	29.53	Soft ripe	27.83	27.20	28.00
Ripe—Pulp	26.00	25.45	26.50	Pears (unknown Japanese variety)	29.39	29.34	29.58
Blackberries:							
Black varieties	29.15	28.73	29.42	Japanese persimmons (Tanenashi)	28.33	28.07	28.63
White varieties	28.40	28.12	28.63	Plums	28.53	28.20	28.85
Logan (Loganberry)	29.51	29.32	29.75	Raspberries:			
Cherries	27.81	27.56	28.25	Red varieties	30.41	30.12	30.50
Cranberries	27.16	26.28	26.93	Black varieties	28.76	28.24	28.79
Currants	30.21	30.18	30.25	Strawberries	29.93	29.56	30.13
Gooseberries	28.91	28.70	29.18	Chestnuts (Italian)	23.80	23.00	24.20
Grapes:				Walnuts (Persian or so-called English)	20.00	19.80	22.10
American	28.16	27.85	28.37				
European	24.60	24.35	24.78				

TABLE 88.—AVERAGE AND EXTREME FREEZING POINTS OF POTATOES SWEET POTATOES, TOMATOES, AND OTHER VEGETABLES.

R. C. WRIGHT and G. F. TAYLOR, U. S. Dept. Agriculture

Kind and variety	Temperatures (° F.) Average	Extremes Minimum	Extremes Maximum
Potatoes:			
Triumph	29.20	29.00	29.33
Early Prospect	28.80	28.72	29.30
Irish Cobbler	29.67	29.60	29.72
First Early	29.00	28.88	29.00
First Early Standard	28.97	28.74	29.12
Ehnola	29.17	29.01	29.30
Spaulding No. 4	29.33	29.21	29.32
Green Mountain	28.50	28.38	28.55
Gold Coin	28.63	28.40	28.70
Rural New Yorker	28.70	28.46	28.75
Russet Rural	28.32	28.30	28.48
U. S. Seedling No. 38774	28.77	28.65	28.83
Up-to-date	29.10	29.19	29.10
Producer	28.70	28.73	28.79
Oregon White Rose	28.71	28.60	28.80
British Queen	29.27	29.22	29.30
Garnet Chile	28.16	28.00	28.28
American Giant	29.64	29.48	29.68
Average	28.92	28.80	29.02
Sweet potatoes:			
Big Stem	28.05	27.48	28.72
Dooley	28.46	27.93	28.91
Early Carolina	28.59	28.40	28.96
Georgia	28.05	27.79	28.58
Gold Skin	28.47	28.21	28.63
Improved Big Stem	28.76	28.26	29.00
Miles	28.34	28.16	28.54
Nancy Hall	28.10	27.54	28.35
Mullihan	27.64	27.46	27.93
Pierson	28.68	28.02	28.72
Porto Rico	28.31	27.87	28.68
Pumpkin	28.98	28.68	29.09
Red Brazil	28.40	28.30	28.63
Red Bermuda	28.17	27.98	28.63
Red Jersey	28.52	28.30	28.77
Southern Queen	28.56	28.25	28.82
Triumph	28.43	28.26	28.72
Yellow Belmont	28.57	28.49	28.82
Yellow Jersey	28.97	28.26	29.05
Yellow Strasburg	28.72	28.30	29.00
Average	28.44	28.10	28.72
Tomatoes (ripe):			
Bonny Best	30.60	30.48	30.68
Olney Special	30.59	30.34	30.67
Earliana	30.52	30.43	30.77
John Baer	30.57	30.24	30.90
Landreth	30.45	30.34	30.72
Early Michigan	30.67	30.19	30.85
Marvel	30.03	29.90	30.38
Bloomdale	29.99	29.90	30.63
Red Rock	30.55	30.48	30.62
Trucker's Favorite	30.06	....	....
New Glory	29.78	29.63	30.38
Tomatoes (ripe)—Contd.			
Stone	30.31	30.10	30.58
Greater Baltimore	30.62	30.20	30.81
Columbia	30.31	30.29	30.77
Delaware Beauty	30.02	29.95	30.33
Livingston's Globe	30.58	30.32	30.88
Livingston's Acme	30.46	30.41	30.74
Greenhouse varieties—			
Carter's Sunrise	30.58	30.06	30.85
Stirling Castle	30.54	30.41	30.60
Average	30.38	30.20	30.67
Tomatoes (green):			
Bonny Best	30.57	30.38	30.83
Earliana	30.21	29.77	30.58
John Baer	30.53	30.48	30.58
Early Michigan	30.70	30.53	30.77
Red Rock	30.58	30.34	30.67
Stone	30.15	30.10	30.38
Greenhouse varieties—			
Carter's Sunrise	30.29	30.20	30.59
Stirling Castle	30.11	29.90	30.15
Average	30.40	30.21	30.57
Sweet corn:			
Crosby	29.07	28.82	29.43
Country Gentleman	29.11	28.63	29.43
Howling Mob	28.00	27.89	28.16
Golden Bantam	29.61	29.25	29.85
Average	28.95	28.65	29.22
Onions:			
Yellow Danvers	30.10	29.61	30.17
White Globe	30.20	29.75	30.41
Texas Bermuda	29.98	29.71	30.13
Average	30.09	29.69	30.24
Lettuce:			
May Queen	30.49	30.38	30.60
Way Ahead	31.54	31.25	31.77
Prize Head	31.57	31.45	31.77
Average	31.20	31.03	31.38
Carrots:			
Danvers	29.61	29.43	29.66
Chantenay	29.53	29.42	29.70
Average	29.57	29.42	29.68
Peas:			
Early Alaska	28.93	28.26	29.19
Horsford's Market Garden	30.93	30.73	30.99
Laxtonian	30.23	30.03	30.56
Average	30.03	29.67	30.25

QUESTIONS ON CHAPTER XV.

1. What are the most important items which make up refrigerating load on a cold storage plant?

2. How is the heat required to cool the materials calculated?

3. In the consideration of the heat transmitted by the insulation of cold storage warehouses, what factors combine to determine the heat transmission coefficient to be used in estimating the loss of refrigeration through the insulation?

4. How may the refrigeration requirements for the ventilation losses and the heat generated in the room be estimated?

5. How do the temperatures required in cold storage warehouses affect the selection of the refrigerant and the refrigerating system to be used for cooling the warehouse?

6. What are the principal factors that determine the suction and condenser pressure in refrigerating systems to be used in cold storage warehouses?

7. Why are the compound compression system and the ammonia absorption system generally used in the larger cold storage plants?

8. Determine the refrigeration requirements for the cold storage building shown in Figs. 169 and 170, when the specific heat of the apples is taken as 0.86 and when the filling period for the building is two weeks.

9. What would be the total refrigeration requirements for the plant described in Problem No. 8 when the water for making 50 tons of ice per day is delivered to the refrigerating system at a temperature of 70° F.? Approximately what suction pressure should be used for the ammonia compressor?

10. Determine the size of the ammonia compressor cylinders to refrigerate the cold storage plant described in Problems No. 8 and 9, when slow-speed horizontal double-acting ammonia compressors are used and when the condensing pressure is 185 lbs. gauge.

CHAPTER XVI.

COOLING AND CONDITIONING OF AIR.

Mechanical Cooling of Air.—One of the more modern applications of mechanical refrigeration consists of the cooling of air by mechanical means. Cooled and properly conditioned air is a necessity in many industrial plants. This is especially true of plants which are devoted to the manufacture of such commodities as munitions, films, chewing gum, bread, rayon, textiles, certain drugs, candies, etc. In addition to these applications air that has been cooled and conditioned is required in hotels, restaurants, theaters, auditoriums, hospitals, stores, offices, etc. Air that has been cooled to a low temperature in order to remove the moisture as much as possible is used in blast furnaces. The use of such air in blast furnaces results in a considerable saving of fuel.

In a similar manner, cooled air is used for the transmission of refrigeration. The precooling of fruit, vegetables, etc., before shipping is an example of this kind of application. Also, cold air may be used for producing the refrigerating effect in cold storage rooms directly. In cold storage warehouses, proper temperatures may be maintained by the forced circulation of cold air. Since the use of mechanically cooled air is becoming quite extensive, as indicated by the number of industrial applications, it is opportune to note the principles underlying the cooling and conditioning of air, as well as the methods and apparatus which are used to produce the cooling effect.

Properties of Air.—In order to thoroughly understand the principles underlying the cooling of air, one must have a good conception of the physical properties of air. The following treatment of the physical properties of air is based upon fundamental principles and methods of treatment which have been presented previously by Carrier, Gebhardt, and others. Air may be considered a mechanical mixture of various gases, such as oxygen, nitrogen, carbonic acid gas, water vapor, etc. Nitrogen and oxygen are the principal components of air and are in dry air in the following proportion:

	By volume	By weight
Oxygen	20.9	23.1
Nitrogen	79.1	76.9

The amount of carbonic acid gas that will be found in air depends upon the locality, the purity of air, etc., and will vary from 0.03 to 0.30 per cent. The amount of water vapor will depend upon the relative temperature and the humidity of the air, and for atmospheric air will vary from 0 to 4 per cent of the total weight. In addition to the foregoing components of air, there are several other gases present, but these are present only in very small quantities.

Perfectly dry air is air which contains no water vapor at all, in which case the air is made up principally of the nitrogen and oxygen components. It is evident that perfectly dry air does not exist at any time in nature, since there is always evaporation of water from the earth's surfaces, which causes the atmospheric air to contain more or less water vapor. However, it is possible to produce dry air by artificial means, so that it is desirable to note the relation of the physical properties. The relation of the pressure, volume, and temperature of dry air may be expressed by the following formula:

$$P_a V_a = \left\{\frac{53.35}{144}\right\} T_a$$

$$\text{or } P_a V_a = 0.37\, T_a$$

where P_a = absolute pressure of the dry air in lbs. per square inch

V_a = volume of the dry air, in cubic feet per lb.

T_a = absolute temperature in degrees F.

The specific heat of dry air at constant pressure varies slightly with the relative intensity of the temperature. The specific heat of dry air may be expressed in a formula as follows:

$$C_{pa} = 0.2411 + 0.0000045\,(t_1 + t_2)$$

where C_{pa} = mean specific heat of dry air at a constant pressure between the temperatures of t_1 and t_2

After the specific heat has been determined and if the range of temperature is known, the amount of heat to be added or extracted from the air in heating or cooling it may be expressed as follows:

$$H_a = C_{pa}\,(t_2 - t_1)$$

where H_a = heat required for cooling or heating

t_2 = higher temperature

t_1 = lower temperature

It is further evident that if the volume of one pound of dry air is represented by V_a the weight of one cubic foot of dry air may be found as follows:

$$w_a = \frac{1}{V_a}$$

where w_a = weight of cubic foot in pounds

Example 1.—Dry air has a temperature of 70° F. Under the normal atmospheric pressure of 14.7 pounds per square inch, it is desired to find the volume per pound, the weight per cubic foot, the mean specific heat between 0° and 70°, and the heat content in Btu. per pound above 0° F.:

$$V_a = \frac{0.37\,T_a}{P_a} = \frac{0.37\,(70 + 459.6)}{14.7}$$
$$= 13.3 \text{ cu. ft. per lb.}$$
$$w_a = \frac{1}{13.3} = 0.075$$
$$C_{pa} = 0.2411 + 0.0000045\,(0 + 70)$$
$$= 0.2414$$
$$H_a = 0.2414\,(70 - 0)$$
$$= 16.9 \text{ Btu.}$$

Saturated Air.—Air is said to be saturated with water vapor when it contains or has mixed with it the maximum amount which is possible at a given temperature. The amount of water vapor that may be present in the air depends upon the temperature and volume of the air. If water is placed in a vacuum, the water will continue to evaporate until the pressure rises to that of the vapor pressure of the water at the corresponding temperature. In a similar manner, if water is placed in a vessel which contains air, it will evaporate until the vapor pressure has risen to that which corresponds to the temperature of the air. Also, according to Dalton's law, each gas or vapor present in the vessel at the given temperature will exert the same amount of pressure as it would have exerted by itself at the same temperature, so that the total pressure would be made up of the sum of the partial pressures exerted by each constituent of the air.

From the foregoing, it will be observed that in order for the air to be saturated it must contain the saturated vapor of water which has a pressure corresponding to the temperature of the air. The water vapor at this temperature and pressure would be at the evaporating or condensing point, so that the extraction of heat from the air and vapor would produce condensation, or the addition of heat to the mixture would cause the water vapor in the air to become superheated, providing the pressure of the air remains constant. According to Dalton's law, the relation of the pressure of the air and the pressure of the vapor may be expressed as follows:

$$P_a + P_v = P_m$$

where P_a = absolute pressure of dry air in the mixture, lbs. per square inch

P_v = absolute pressure of the saturated water vapor in the mixture, lbs. per square inch

P_m = absolute pressure of the mixture, lbs. per square inch

TABLE 89.—PROPERTIES OF LOW PRESSURE STEAM.
(Wheeler Condenser and Engineering Co.)

Temperature Fahr. t	Vacuum in Inches of Mercury referred to a 30" Bar. (Mercury at 58.4° F.)	Pressure Lbs. per Sq. In. Absolute P	Pressure Inches of Mercury, with Mercury at 32° F.	Specific Volume Cu. Ft. per Lb. v	Heat of the Liquid h	Total Heat of Steam H	Entropy of Water n	Entropy of Steam N
32°	29.8191	0.0886	0 1804	3294	0 00	1073 4	0 0000	2.1832
40	29.7516	0.1217	0.2477	2438.	8.05	1076.9	0.0162	2.1556
50°	29.6365	0.1780	0 3625	1702.	18.08	1081 4	0.0361	2.1226
51	29.6228	0.1848	0 3762	1643.	19.08	1081.9	0.0381	2.1195
52	29.6087	0.1917	0.3903	1586.	20.08	1082.3	0.0401	2.1164
53	29.5940	0.1989	0.4049	1532.	21.08	1082.7	0.0420	2.1132
54	29.5788	0.2063	0.4201	1480.	22.08	1083.2	0.0440	2.1100
55°	29.5631	0.2140	0.4357	1430.	23 08	1083.6	0.0459	2.1068
56	29.5470	0.2219	0.4518	1381.	24.08	1084.1	0.0478	2.1037
57	29.5303	0.2301	0.4684	1335.	25.08	1084.5	0.0498	2.1006
58	29.5131	0.2385	0.4856	1291.	26.08	1085.0	0.0517	2.0975
59	29.4953	0.2472	0.5034	1249.	27.08	1085.4	0.0536	2.0944
60°	29.477	0.2562	0.522	1208.	28.08	1085.9	0.0555	2.0913
61	29.458	0.2654	0.541	1168.	29.08	1086.3	0.0574	2.0882
62	29.439	0.2749	0.560	1130.	30.08	1086.8	0.0593	2.0851
63	29.419	0.2847	0.580	1093.	31.07	1087.2	0.0612	2.0821
64	29.398	0.2949	0.601	1058.	32.07	1087.6	0.0631	2.0791
65°	29.376	0.3054	0.622	1024.	33.07	1088.1	0.0650	2.0760
66	29.354	0.3161	0.644	991.	34.07	1088.5	0.0669	2.0731
67	29.331	0.3272	0.667	959.	35.07	1089.0	0.0688	2.0701
68	29.308	0.3386	0.690	928.	36.07	1089.4	0.0707	2.0672
69	29.284	0.3504	0.714	899.	37.06	1089.9	0.0726	2.0642
70°	29.259	0.3626	0.739	871.	38.06	1090.3	0.0745	2.0613
71	29.234	0.3751	0.764	843.	39.06	1090.8	0.0764	2.0585
72	29.208	0.3880	0.790	817	40.05	1091.2	0.0783	2.0556
73	29.181	0.4012	0.817	792.	41.05	1091.6	0.0802	2.0528
74	29.153	0.4148	0.845	767.	42.05	1092.1	0.0821	2.0499
75°	29.125	0.4288	0.873	743.	43.05	1092.5	0.0840	2.0471
76	29.095	0.4432	0.903	720.	44.04	1093.0	0.0858	2.0443

The magnitude of the pressures of saturated water vapor or steam at temperatures ranging from 32° to 212° may be taken from Table 89. The relation of the pressures, volumes, and temperatures of dry air saturated with water vapor may be stated as follows:

$$P_a = P_m - P_v$$

$$(P_m - P_v)\, V_a = \left[\frac{53.35}{144}\right] T_a$$

$$(P_m - P_v)\, V_a = 0.37\, T_a$$

TABLE 89.—PROPERTIES OF LOW PRESSURE STEAM.—(Continued.)

Temperature Fahr. t	Vacuum in Inches of Mercury referred to a 30" Bar. (Mercury at 58.4° F.)	Pressure Lbs. per Sq. In. Absolute p	Pressure Inches of Mercury, with Mercury at 32° F.	Specific Volume Cu. Ft. per Lb. v	Heat of the Liquid h	Total Heat of Steam H	Entropy of Water n	Entropy of Steam N
77°	29.065	0.4581	0.933	698.	45.04	1093.4	0.0876	2.0414
78	29.034	0.4735	0.964	677.	46.04	1093.9	0.0895	2.0386
79	29.002	0.4893	0.996	657.	47.04	1094.3	0.0913	2.0358
80°	28.968	0.505	1.029	636.8	48.03	1094.8	0.0932	2.0330
81	28.934	0.522	1.063	617.5	49.03	1095.2	0.0950	2.0302
82	28.899	0.539	1.098	598.7	50.03	1095.6	0.0969	2.0275
83	28.863	0.557	1.134	580.5	51.02	1096.1	0.0987	2.0247
84	28.826	0.575	1.171	562.9	52.02	1096.5	0.1005	2.0220
85°	28.788	0.594	1.209	545.9	53.02	1097.0	0.1023	2.0192
86	28.749	0.613	1.248	529.5	54.01	1097.4	0.1041	2.0165
87	28.708	0.633	1.289	513.7	55.01	1097.9	0.1060	2.0139
88	28.666	0.654	1.331	498.4	56.01	1098.3	0.1078	2.0112
89	28.624	0.675	1.373	483.6	57.00	1098.7	0.1096	2.0085
90°	28.580	0.696	1.417	469.3	58.00	1099.2	0.1114	2.0058
91	28.535	0.718	1.462	455.5	59.00	1099.6	0.1133	2.0033
92	28.489	0.741	1.508	442.2	60.00	1100.1	0.1151	2.0007
93	28.441	0.765	1.556	429.4	60.99	1100.5	0.1169	1.9981
94	28.392	0.789	1.605	417.0	61.99	1101.0	0.1187	1.9954
95°	28.341	0.813	1.655	405.0	62.99	1101.4	0.1205	1.9928
96	28.290	0.838	1.706	393.4	63.98	1101.8	0.1223	1.9903
97	28.237	0.864	1.759	382.2	64.98	1102.3	0.1241	1.9877
98	28.183	0.891	1.813	371.4	65.98	1102.8	0.1259	1.9851
99	28.127	0.918	1.869	360.9	66.97	1103.2	0.1277	1.9826
100°	28.070	0.946	1.926	350.8	67.97	1103.6	0.1295	1.9800
101	28.011	0.975	1.985	341.0	68.97	1104.0	0.1313	1.9776
102	27.951	1.005	2.045	331.5	69.96	1104.5	0.1330	1.9750
103	27.889	1.035	2.107	322.2	70.96	1104.9	0.1347	1.9724
104°	27.825	1.066	2.171	313.3	71.96	1105.3	0.1365	1.9700
105	27.759	1.098	2.236	304.7	72.95	1105.8	0.1383	1.9675

The relation of the weight per cubic foot and the volume of one pound can be expressed as follows:

$$w_a = \frac{1}{V_a}$$

The total weight of the mixture in pounds per cubic foot can be expressed as follows:

$$w_m = w_a + w_v$$

where w_m = weight of mixture, lbs. per cu. ft.

w_v = weight of vapor in one cu. ft. of the mixture

TABLE 89.—PROPERTIES OF LOW PRESSURE STEAM.—(Continued.)

Temperature Fahr. t	Vacuum in Inches of Mercury referred to a 30″ Bar. (Mercury at 58.4° F.)	Pressure Lbs. per Sq. In. Absolute P	Pressure Inches of Mercury, with Mercury at 32° F.	Specific Volume Cu. Ft. per Lb. v	Heat of the Liquid h	Total Heat of Steam H	Entropy of Water n	Entropy of Steam N
106°	27.692	1.131	2.303	296.4	73.95	1106.2	0.1401	1.9651
107	27.623	1.165	2.372	288.3	74.95	1106.7	0.1418	1.9626
108°	27.550	1.199	2.443	280.5	75.95	1107.1	0.1436	1.9602
109	27.478	1.235	2.515	272.9	76.94	1107.5	0.1454	1.9578
110°	27.404	1.271	2.589	265.5	77.94	1108.0	0.1471	1.9553
111	27.328	1.308	2.665	258.3	78.94	1108.4	0.1489	1.9530
112	27.250	1.346	2.740	251.4	79.93	1108.8	0.1506	1.9506
113	27.170	1.386	2.822	244.7	80.93	1109.3	0.1524	1.9483
114	27.088	1.426	2.904	238.2	81.93	1109.7	0.1541	1.9458
115°	27.005	1.467	2.987	231.9	82.92	1110.2	0.1559	1.9435
116	26.919	1.509	3.073	225.8	83.92	1110.6	0.1576	1.9412
117	26.830	1.553	3.161	219.9	84.92	1111.0	0.1594	1.9389
118	26.739	1.597	3.252	214.1	85.92	1111.5	0.1611	1.9366
119	26.647	1.642	3.344	208.5	86.91	1111.9	0.1628	1.9343
120°	26.553	1.689	3.438	203.1	87.91	1112.3	0.1645	1.9319
121	26.456	1.736	3.535	197.9	88.91	1112.8	0.1662	1.9296
122	26.355	1.785	3.635	192.8	89.91	1113.2	0.1679	1.9273
123	26.253	1.835	3.737	187.9	90.90	1113.6	0.1696	1.9251
124	26.149	1.886	3.841	183.1	91.90	1114.1	0.1713	1.9228
125°	26.040	1.938	3.948	178.4	92.90	1114.5	0.1730	1.9205
126	25.931	1.992	4.057	173.9	93.90	1115.0	0.1747	1.9183
127	25.820	2.047	4.168	169.6	94.89	1115.4	0.1764	1.9161
128	25.706	2.103	4.282	165.3	95.89	1115.8	0.1781	1.9139
129	25.589	2.160	4.399	161.1	96.89	1116.2	0.1799	1.9117
130°	25.48	2.219	4.52	157.1	97.89	1116.7	0.1816	1.9095
131	25.35	2.279	4.64	153.2	98.89	1117.1	0.1833	1.9073
132	25.23	2.340	4.76	149.4	99.88	1117.5	0.1849	1.9051
133	25.10	2.403	4.89	145.8	100.88	1118.0	0.1866	1.9030
134	24.97	2.467	5.02	142.2	101.88	1118.4	0.1883	1.9008

The values for the weights of a cubic foot of the water vapor at various temperatures may be obtained from Table 89. This table gives the specific volume in cubic feet per pound for saturated steam at various temperatures. By taking the reciprocals of these volumes, the corresponding weights per cubic foot may be obtained. From the foregoing, it will be observed that it is possible to calculate the amount of water vapor required to saturate a given volume of air or a given weight of air, when the temperatures and pressures are known. This may be expressed in a formula as follows:

TABLE 89.—PROPERTIES OF LOW PRESSURE STEAM.—(Concluded)

Temperature Fahr. t	Vacuum in Inches of Mercury referred to a 30" Bar. (Mercury at 58.4° F.)	Pressure p Lbs. per Sq. In. Absolute	Pressure Inches of Mercury, with Mercury at 32° F.	Specific Volume Cu. Ft. per Lb. v	Heat of the Liquid h	Total Heat of Steam H	Entropy of Water n	Entropy of Steam N
135°	21.83	2.533	5.16	138.7	102.88	1118.8	0.1900	1.8986
140	24.11	2.885	5.88	122.8	107.87	1121.0	0.1984	1.8880
150°	22.42	3.714	7.57	96.9	117.86	1125.3	0.2149	1.8674
160	20.32	4.737	9.65	77.2	127.86	1129.5	0.2311	1.8476
170	17.77	5.992	12.20	62.0	137.87	1133.7	0.2470	1.8286
180	14.67	7.51	15.29	50.15	147.88	1137.8	0.2628	1.8104
190	10.93	9.34	19.02	40.91	157.91	1141.8	0.2783	1.7929
200°	6.47	11.52	23.47	33.60	167.94	1145.8	0.2937	1.7761
210	1.16	14.13	28.76	27.80	177.99	1149.6	0.3087	1.7597
212	0.00	14.70	29.92	26.79	180.00	1150.4	0.3118	1.7565

NOTE

In the foregoing table three methods of expressing a pressure are used; thus in the last line, opposite 212° F. we have, vacuum = 0.00; absolute pressure in lbs. per sq. inch = 14.7 and in inches of mercury at 32° F. = 29.92. These are measures of exactly the same pressure, expressed in different terms, and may be used as a basis for making other transformations besides those given in this book. 1 lb. per sq. inch = 2.0355 in. mercury at 32° F. mercury temperature, and 1 in. mercury = .49 lbs. per sq. inch. Also, 0.00 vacuum referred to a 30-in. barometer = 30.0" absolute pressure of mercury at 58.4° F. temperature of mercury, or 29.92" absolute pressure of mercury at 32° F. temperature of mercury.

$$w = V_a \times w_v$$

where w = weight of water vapor to saturate one pound of dry air

Furthermore, it is evident that the heat content of the air at a given temperature will depend upon two factors: First, the heat content of the dry air above a given temperature; second, the heat in the saturated steam. The heat in the saturated steam may be taken to be that due to the latent heat of vaporization of the water vapor, neglecting the heat of the liquid, which is a very small quantity. This may be expressed symbolically as follows for the heat measure above 0° F.:

$$H_{sa} = C_{pa} \times t_a + r_v \times w$$

where H_{sa} = heat content of saturated air, B.t.u. per lb.

t_a = temperature of the mixture of air and steam in degrees F.

r_v = latent heat of vaporization of the water vapor at the temperature of the mixture, t_a

The values of the latent heat of vaporization, r_v, may be taken from Table 89 by remembering that the latent heat of vaporization is equal to the total heat of the steam minus the heat of the liquid, or, in symbols:

$$r_v = H - h$$
where H = total heat of steam from Table 89
h = heat of liquid from Table 89

The foregoing formulas were used for the calculation of Table 90, Properties of Saturated Air.

Example 1.—Air is saturated with water vapor at a temperature of 70° F., and the pressure is 14.7 lbs. per sq. in. It is desired to determine the pressure of the vapor, the pressure of the dry air in the mixture, the volume of one pound of the mixture, the relative weights of dry air and water vapor in a cubic foot, the weight of the moisture to saturate one pound of air, and the heat content above 0° of one pound of air:

Pressure of water, vapor in mixture from Table 89,
$$P_v = 0.363 \text{ lbs. per sq. in.}$$
Pressure of dry air in the mixture,
$$P_a = P_m - P_v$$
$$= 14.700 - 0.363$$
$$= 14.337 \text{ lbs. per sq. in.}$$
Volume of one lb. of the mixture,
$$V_a = \frac{0.37\ T_a}{P_m - P_v} = \frac{0.37\ (70 + 459.6)}{14.337}$$
$$V_a = 13.69$$
Weight of one cu. ft. of mixture,
$$w_a = \frac{1}{V_a} = \frac{1}{13.69} = 0.0731$$
Weight of water vapor to saturate one lb. of dry air,
$$w = V_a \times w_v$$
$$w^v \text{ from Table LXXV} = \frac{1}{871} = 0.001148$$
$$w = 13.69 \times 0.001148$$
$$= 0.01578 \text{ lbs. per lb. of dry air}$$
Heat content of dry air and vapor about 0° F., neglecting the heat of the liquid,
$$H_{sa} = C_{pa} \times t_a + r_v \times w$$
From Table 89, latent heat of evaporation,
$$r = 1090.3 - 38.06 = 1052.24$$
$$H_{sa} = 16.9 + (0.001148 \times 1052.24)$$
$$= 16.9 + 16.61 = 33.51$$

Partially Saturated Air.—As previously indicated, air is saturated when it contains the saturated vapor of water, and it was also observed that the weight of the vapor per unit volume corresponds to the temperature of the saturated steam at the temperature of the mixture. Also, it was pointed out that the air does not affect in any manner the

TABLE 90.—PROPERTIES OF SATURATED AIR. MIXTURE OF AIR SATURATED WITH WATER VAPOR.

G. F. Gebhardt

Temperature, Degrees Fahr.	Weight of Water Necessary to Saturate 100 Lb. of Dry Air.	Volume of One Pound of Dry Air + Vapor to Saturate it, Cubic Feet.	Heat Content per Pound of Dry Air, B.t.u.	Latent Heat of Vapor in One Lb. of Dry Air Saturated with Vapor, B.t.u.	Heat Content of One Lb. of Dry Air Saturated with Vapor, B.t.u.
0	0.078	11.59	0.000	0.964	0.964
10	0.131	11.86	2.411	1.608	4.019
20	0.214	12.13	4.823	2.623	7.446
30	0.344	12.41	7.234	4.195	11.429
32	0.378	12.47	7.716	4.058	11.783
35	0.427	12.55	8.44	4.57	13.02
40	0.520	12.70	9.65	5.56	15.21
45	0.632	12.85	10.86	6.73	17.59
50	0.764	13.00	12.07	8.12	20.19
55	0.920	13.16	13.28	9.76	23.04
60	1.105	13.33	14.48	11.69	26.18
62	1.188	13.40	14.97	12.12	26.84
65	1.323	13.50	15.69	13.96	29.65
70	1.578	13.69	16.90	16.61	33.51
72	1.692	13.76	17.38	17.79	35.17
75	1.877	13.88	18.11	19.71	37.81
80	2.226	14.09	19.32	23.31	42.64
85	2.634	14.31	20.53	27.51	48.04
90	3.109	14.55	21.74	32.39	54.13
95	3.662	14.80	22.95	38.06	61.01
100	4.305	15.08	24.16	44.63	68.79
105	5.05	15.39	25.37	52.26	77.63
110	5.93	15.73	26.58	61.11	87.69
115	6.94	16.10	27.79	71.40	99.10
120	8.13	16.52	29.00	83.37	112.37
125	9.53	16.99	30.21	97.33	127.54
130	11.14	17.53	31.42	113.64	145.06
135	13.05	18.13	32.63	132.71	165.34
140	15.32	18.84	33.85	155.37	189.22
145	18.00	19.64	35.06	182.05	217.10
150	21.22	20.60	36.27	214.03	250.30
155	25.11	21.73	37.48	252.61	290.10
160	29.87	23.09	38.69	299.55	338.20
165	35.77	24.75	39.91	357.75	397.70
170	43.24	26.84	41.12	431.20	472.30
175	52.90	29.51	42.33	526.0	568.30
180	65.77	33.04	43.55	651.9	695.50
185	83.59	37.89	44.76	826.1	870.90
190	109.80	45.00	45.97	....	....
195	191.00	56.20	47.20	....	....
200	229.50	77.24	48.40	....	....
205	419.00	....	49.62	....	....
210	....	....	50.83	....	....
212	....	....	51.39	....	....

amount of the water vapor in a given volume, but only affects the total pressure and weight of the mixture. If a unit volume of air contains only part of the weight of the vapor corresponding to the saturation point, it is said to be partially saturated, and the relative amount of the water vapor present is known as the relative humidity. In other words, the relative humidity is the ratio of the actual amount of moisture contained in a unit volume of the mixture to the amount which the unit volume would hold at the same temperature if it were saturated.

The actual or absolute humidity is the weight of water vapor in a given volume of air and is usually expressed in pounds of water vapor per cubic foot of air. If the relative humidity is below 100 per cent, it is evident that the water vapor must be in a superheated state, because this is the only way that the relative weight of the water vapor in a unit volume of air could be decreased. This is due to the fact that the temperature of the vapor in the mixture is higher than the temperature of the saturated vapor corresponding to the pressure of the vapor in the mixture. It will be also observed that atmospheric air, in general, contains superheated water vapor or steam, and that it contains saturated steam only when the moisture is condensing out of the atmosphere; that is, it is raining.

From the foregoing, it is evident that if partially saturated air is cooled at constant pressure, there will be a temperature at which the water vapor will become saturated and that the further reduction of temperature will cause condensation. This condensing point is known as the dew point and is the temperature at which saturation is obtained. The relative humidity and the dew point of atmospheric air are ordinarily determined by use of an instrument which is called the psychrometer. The sling psychrometer consists of two thermometers which are suitably mounted and attached to a handle so that they may be rotated. Wet bulb thermometer, as the name indicates, is a bulb which is covered with a piece of soft cloth or wick which is kept moist with water. The dry bulb is exposed directly to the air. When the sling psychrometer is rotated or whirled at a rate of 150 to 200 revolutions per minute, evaporation takes place on the wet bulb thermometer, so that a depressed temperature reading is secured. By means of the temperature readings on the wet and dry bulb thermometers, it is possible to determine the relative humidity, the dew point, and the amount of water vapor in the air. The relation of the dew point, relative humidity, moisture content and temperature is shown by Table 91.

If the air is saturated with water vapor, it is evident that no evaporation will take place on the wet bulb thermometer, and that the readings of the wet and dry bulb thermometers will be the same. As the relative humidity is decreased, there will be a correspondingly in-

TABLE 91.—RELATIVE HUMIDITY, DEW POINTS AND GRAINS OF MOISTURE PER CUBIC FOOT FOR VARIOUS DRY AND WET BULB TEMPERATURES OF THE SLING PSYCHROMETER-BAROMETRIC PRESSURE, 30 INCHES.

Carrier Corporation

WET BULB DEPRESSION

Dry Bulb Temp.	Grains at Saturation	1			2			3			4		
		%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins
0	.48	67	-7	.32	33	-20	16						
2	.53	70	-5	.37	39	-15	21	9	-40	05			
4	58	72	-2	42	44	-11	26	17	-28	10			
6	.64	74	0	47	49	-8	.32	23	-21	15			
8	.70	76	3	.54	53	-5	.37	29	-15	20	6	-42	04
10	.78	78	5	.61	56	-2	44	34	-10	26	13	-27	11
12	.86	80	7	69	59	2	51	39	-6	.33	19	-19	16
14	.94	81	10	.76	62	5	.58	44	-2	.41	26	-12	25
16	1.03	82	12	.85	65	7	65	48	+1	50	31	-7	32
18	1.13	84	14	.95	68	10	72	52	5	59	36	-2	41
20	1.24	85	16	1.1	70	12	86	55	8	68	40	+2	49
22	1.36	86	19	1.17	71	15	96	58	11	.79	44	5	60
24	1.48	87	21	1.30	73	17	1.1	60	13	.89	47	9	70
26	1.62	87	23	1.41	75	20	1.22	63	16	1.02	51	12	.83
28	1.77	88	25	1.56	76	22	1.37	65	19	1.15	54	15	96
30	1.94	89	27	1.72	78	25	1.5	67	21	1.30	56	18	1.08
32	2.11	89	30	1.88	79	27	1.67	69	24	1.46	59	21	1.25
34	2.28	90	32	2.05	81	29	1.85	71	26	1.62	62	23	1.41
36	2.46	91	34	2.24	82	31	2.02	73	29	1.79	64	26	1.57
38	2.65	91	36	2.41	83	33	2.20	75	31	1.98	66	28	1.74
40	2.85	92	38	2.62	83	35	2.36	75	33	2.14	68	30	1.94
42	3.06	92	40	2.82	85	38	2.61	77	35	2.36	69	33	2.12
44	3.29	93	42	3.06	85	40	2.80	78	37	2.57	71	35	2.34
46	3.54	93	44	3.29	86	42	3.04	79	40	2.80	72	37	2.54
48	3.80	93	46	3.53	86	44	3.27	79	42	3.00	73	40	2.77
50	4.08	93	48	3.79	87	46	3.55	80	44	3.26	74	42	3.02
52	4.37	94	50	4.11	87	48	3.80	81	46	3.54	75	44	3.28
54	4.69	94	52	4.40	88	50	4.13	82	48	3.84	76	46	3.56
56	5.02	94	54	4.71	88	53	4.41	82	51	4.11	76	49	3.81
58	5.37	94	56	5.05	88	55	4.73	83	53	4.46	77	51	4.14
59	5.56	94	57	5.22	89	56	4.95	83	54	4.61	78	52	4.33
60	5.75	94	58	5.40	89	57	5.11	83	55	4.77	78	53	4.48
61	5.94	94	59	5.58	89	58	5.29	84	56	4.99	78	54	4.64
62	6.14	94	60	5.77	89	59	5.46	84	57	5.16	79	55	4.85
63	6.35	95	61	6.03	89	60	5.65	84	58	5.23	79	56	5.01
64	6.56	95	62	6.24	90	61	5.91	84	59	5.51	79	57	5.19

Dry Bulb Temp.	5			6			7			8			9		
	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins
0															
2															
4															
6															
8															
10															
12															
14	8	-33	08												
16	14	-20	15												
18	20	-13	23	5	-37	06									
20	26	-7	32	12	-21	15									
22	31	-2	42	17	-12	23	4	-36	.05						
24	35	+2	52	22	-6	33	10	-20	.15						
26	39	7	63	27	-1	44	16	-11	26	4	-32	.07			
28	43	10	76	32	+4	52	21	-4	37	10	-17	18			
30	46	14	89	36	8	.70	26	+2	.50	16	-7	.31	6	-25	.12
32	49	17	1.03	39	12	0.82	30	7	0.63	20	-1	0.42	11	-12	0.23
34	52	20	1.19	43	16	0.98	34	11	0.78	25	5	0.57	16	-3	0.42
36	55	23	1.35	46	19	1.13	38	15	0.93	29	10	0.71	21	3	0.52
38	58	25	1.54	50	22	1.32	42	18	1.11	33	14	0.87	25	8	0.66
40	60	28	1.71	52	25	1.48	45	21	1.28	37	18	1.06	29	13	0.83
42	62	30	1.90	55	27	1.69	47	24	1.44	40	21	1.23	33	17	1.01
44	63	32	2.08	56	30	1.85	49	27	1.61	43	24	1.42	36	20	1.19
46	65	35	2.30	58	32	2.05	52	29	1.84	45	27	1.59	39	23	1.38
48	66	37	2.51	60	35	2.28	54	32	2.05	47	29	1.79	41	26	1.56
50	67	40	2.73	61	37	2.49	55	34	2.24	49	32	1.20	43	29	1.75
52	69	42	3.02	63	40	2.75	57	37	2.49	51	34	2.23	46	32	2.01
54	70	44	3.28	64	42	3.00	59	40	2.77	53	37	2.48	48	34	2.25
56	71	47	3.61	65	44	3.26	60	42	3.01	55	40	2.76	50	37	2.51
58	72	49	3.87	66	47	3.55	61	45	3.28	56	42	3.01	51	40	2.74
59	72	50	4.00	67	48	3.72	62	46	3.44	57	44	3.17	52	41	2.89
60	73	51	4.19	68	49	3.91	63	47	3.62	58	45	3.33	53	43	3.04
61	73	52	4.34	68	50	4.04	63	48	3.74	58	46	3.45	54	44	3.21
62	74	53	4.54	69	52	4.24	64	50	3.93	59	47	3.62	54	45	3.32
63	74	55	4.71	69	53	4.38	64	51	4.07	60	49	3.81	55	47	3.49
64	74	56	4.86	70	54	4.59	65	52	4.27	60	50	3.94	56	48	3.68

TABLE 91.—RELATIVE HUMIDITY, DEW POINTS AND GRAINS OF MOISTURE PER CUBIC FOOT FOR VARIOUS DRY AND WET BULB TEMPERATURES OF THE SLING PSYCHROMETER-BAROMETRIC PRESSURE, 30 INCHES.—(Continued.)

		WET BULB DEPRESSION																										
		1			2			3			4			5			6			7			8			9		
Dry Bulb Temp.	Grains at Saturation	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins
65	6.78	95	63	6.44	90	62	6.10	85	60	5.77	80	59	5.43	75	57	5.09	70	55	4.75	66	53	4.48	61	51	4.14	56	49	3.80
66	7.01	95	64	6.66	90	63	6.31	85	61	5.96	80	60	5.61	75	58	5.26	71	56	4.98	66	54	4.63	61	52	4.28	57	50	3.99
67	7.24	95	65	6.88	90	64	6.52	85	62	6.16	80	61	5.79	75	59	5.43	71	57	5.14	66	55	4.78	62	53	4.49	58	52	4.20
68	7.48	95	67	7.10	90	65	6.73	85	63	6.36	80	62	5.98	76	60	5.69	71	58	5.31	67	57	5.01	62	55	4.64	58	53	4.34
69	7.73	95	68	7.34	90	66	6.95	85	64	6.57	81	63	6.26	76	61	5.87	72	59	5.56	67	58	5.18	63	56	4.87	59	54	4.56
70	7.98	95	69	7.58	90	67	7.18	86	65	6.86	81	64	6.46	77	62	6.15	72	61	5.75	68	59	5.43	64	57	5.11	59	55	4.71
71	8.24	95	70	7.83	90	68	7.42	86	67	7.09	81	65	6.67	77	63	6.35	72	62	5.93	68	60	5.60	64	58	5.27	60	56	4.94
72	8.51	95	71	8.08	91	69	7.74	86	68	7.31	82	66	6.97	77	64	6.55	73	63	6.21	69	61	5.87	65	59	5.53	61	58	5.19
73	8.78	95	72	8.34	91	70	7.99	86	69	7.55	82	67	7.20	78	66	6.85	73	64	6.41	69	62	6.06	65	60	5.71	61	59	5.36
74	9.07	95	73	8.61	91	71	8.25	86	70	7.80	82	68	7.43	78	67	7.07	74	65	6.71	69	63	6.26	65	62	5.89	61	60	5.53
75	9.36	96	74	8.89	91	72	8.51	86	71	8.05	82	69	7.67	78	68	7.30	74	66	6.92	70	64	6.59	66	63	6.18	62	61	5.80
76	9.66	96	75	9.27	91	73	8.79	87	72	8.40	82	70	7.92	78	69	7.53	74	67	7.14	70	66	6.76	66	64	6.37	62	62	5.99
77	9.96	96	76	9.56	91	74	9.02	87	73	8.67	83	71	8.27	79	70	7.87	74	68	7.37	71	67	7.07	67	65	6.67	63	63	6.23
78	10.28	96	77	9.87	91	75	9.35	87	74	8.94	83	72	8.53	79	71	8.12	75	69	7.71	71	68	7.30	67	66	6.89	63	64	6.47
79	10.60	96	78	10.18	91	76	9.65	87	75	9.21	83	73	8.80	79	72	8.38	75	70	7.95	71	69	7.53	68	67	7.21	64	66	6.79
80	10.93	96	79	10.50	91	77	9.95	87	76	9.51	83	74	9.08	79	73	8.64	75	72	8.20	72	70	7.87	68	68	7.44	64	67	7.00
81	11.28	96	80	10.82	92	78	10.37	88	77	9.92	84	75	9.47	80	74	9.02	76	73	8.57	72	71	8.12	69	70	7.78	65	68	7.33
82	11.63	96	81	11.16	92	79	10.70	88	78	10.23	84	77	9.77	80	75	9.30	76	74	8.84	72	72	8.37	69	71	8.02	65	69	7.56
83	11.99	96	82	11.51	92	80	11.03	88	79	10.55	84	78	10.07	80	76	9.59	76	75	9.11	73	73	8.75	69	72	8.27	66	70	7.91
84	12.36	96	83	11.86	92	81	11.37	88	80	10.87	84	79	10.38	80	77	9.89	76	76	9.39	73	74	9.02	69	73	8.53	66	71	8.16
85	12.74	96	84	12.22	92	82	11.72	88	81	11.21	84	80	10.70	81	78	10.32	77	77	9.81	73	75	9.30	70	74	8.92	66	72	8.41
86	13.13	96	85	12.60	92	83	12.08	88	82	11.55	84	81	11.03	81	79	10.63	77	78	10.11	73	76	9.58	70	75	9.19	66	73	8.67
87	13.53	96	86	12.99	92	84	12.44	88	83	11.90	85	82	11.50	81	80	10.96	77	79	10.42	74	78	10.01	70	76	9.47	67	75	9.06
88	13.94	96	87	13.38	92	85	12.82	88	84	12.26	85	83	11.85	81	81	11.29	77	80	10.73	74	79	10.31	70	77	9.76	67	76	9.34
89	14.36	96	88	13.99	92	86	13.21	88	85	12.64	85	84	12.21	81	82	11.63	78	81	11.20	74	80	10.63	71	78	10.19	68	77	9.76
90	14.79	96	89	14.20	92	87	13.61	89	86	13.16	85	85	12.57	81	83	11.98	78	82	11.54	74	81	10.94	71	79	10.50	68	78	10.06
92	15.69	96	91	15.06	92	89	14.44	89	88	13.96	85	87	13.34	82	86	12.87	78	84	12.24	75	83	11.77	72	81	11.30	68	80	10.67
94	16.63	96	93	15.97	93	92	15.47	89	90	14.81	85	89	14.14	82	88	13.64	79	86	13.14	75	85	12.48	72	84	11.98	69	82	11.48
96	17.63	96	95	16.92	93	94	16.39	89	92	15.69	86	91	15.16	82	90	14.45	79	88	13.93	76	87	13.40	73	86	12.87	69	84	12.16
98	18.67	96	97	17.92	93	96	17.36	89	94	16.62	86	93	16.06	83	92	15.50	79	90	14.75	76	89	14.19	73	88	13.63	70	87	13.07
100	19.77	96	99	18.98	93	98	18.38	89	96	17.59	86	95	17.00	83	94	16.41	80	93	15.81	77	91	15.22	73	90	14.43	70	89	13.84
104	22.13	97	103	21.46	93	102	20.58	90	100	19.91	87	99	19.25	83	98	18.36	80	97	17.70	77	95	17.04	74	94	16.37	71	93	15.71
108	24.72	97	107	23.98	93	106	22.99	90	104	22.25	87	103	21.51	84	102	20.77	81	101	20.02	78	100	19.28	75	98	18.54	72	97	17.80
112	27.88	97	111	27.02	94	110	26.19	90	109	25.07	87	107	24.24	84	106	23.40	81	105	22.57	79	104	21.60	76	103	20.90	73	101	20.34
116	30.10	97	115	29.20	94	114	28.29	91	113	27.39	88	111	27.21	85	110	26.31	82	109	25.40	79	108	24.50	76	107	23.60	74	105	22.99
120	34.80	97	119	33.76	94	118	32.71	91	117	31.67	88	115	30.62	85	114	29.58	82	113	28.88	80	112	27.84	77	111	26.78	74	110	25.71

TABLE 91.—RELATIVE HUMIDITY, DEW POINTS AND GRAINS OF MOISTURE PER CUBIC FOOT FOR VARIOUS DRY AND WET BULB TEMPERATURES OF THE SLING PSYCHROMETER-BAROMETRIC PRESSURE, 30 INCHES.—(Continued.)

WET BULB DEPRESSION

Dry Bulb Temp.	10			11			12			13		
	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins
32	2	-43	0.04									
34	8	-17	0.18	0								
36	13	-6	0.32	5	-25	0.12						
38	17	1	0.45	10	-10	0.27	2	-36	0.05			
40	22	7	0.63	15	-1	0.43	7	-14	0.20	0		
42	26	12	0.80	19	6	0.58	12	-3	0.37	5	-18	0.15
44	30	16	0.99	23	11	0.76	16	4	0.53	10	-5	0.33
46	32	20	1.13	26	15	0.92	20	10	0.71	14	3	0.50
48	35	23	1.33	29	19	1.11	23	14	0.88	18	9	0.68
50	38	26	1.55	32	22	1.30	27	18	1.10	21	13	0.80
52	40	29	1.75	35	26	1.53	29	22	1.27	24	18	1.05
54	42	32	1.97	37	29	1.73	32	25	1.49	27	22	1.27
56	44	34	2.21	39	32	1.96	34	20	1.71	30	25	1.51
58	46	37	2.47	41	35	2.20	37	32	1.09	32	29	1.72
59	47	39	2.61	42	36	2.33	38	33	2.11	33	30	1.83
60	48	40	2.76	43	38	2.47	39	35	2.25	34	32	1.95
61	49	42	2.97	44	39	2.61	40	36	2.38	35	33	2.08
62	50	43	3.07	45	40	2.76	41	38	2.52	36	35	2.21
63	50	44	3.28	46	42	2.92	42	39	2.67	37	36	2.34
64	51	46	3.35	47	43	3.09	43	41	2.82	38	38	2.49
65	52	47	3.53	48	45	3.26	44	42	2.98	39	40	2.65
66	53	48	3.71	48	46	3.36	44	44	3.08	40	41	2.80
67	53	49	3.84	49	47	3.55	45	45	3.19	41	43	2.97
68	54	51	4.04	50	49	3.74	46	46	3.44	42	44	3.14
69	55	52	4.25	51	50	3.94	47	48	3.63	43	45	3.32
70	55	53	4.39	51	51	4.07	48	49	3.83	44	47	3.51
71	56	54	4.62	52	52	4.29	48	50	3.96	45	48	3.71
72	57	56	4.86	53	54	4.51	49	52	4.17	45	50	3.83
73	57	57	5.01	53	55	4.65	50	53	4.39	46	51	4.04
74	58	58	5.26	54	56	4.90	50	54	4.53	47	52	4.26
75	58	59	5.43	54	57	5.05	51	55	4.77	47	54	4.40
76	59	60	5.70	55	59	5.31	51	57	4.92	48	55	4.64
77	59	62	5.88	56	60	5.58	52	58	5.18	48	56	4.78
78	60	63	6.17	56	61	5.76	53	59	5.45	49	57	5.04
79	60	64	6.36	57	62	6.04	53	60	5.62	50	59	5.30

Dry Bulb Temp.	14			15			16			17			18		
	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins
32															
34															
36															
38															
40															
42															
44	4	-24	0.13												
46	8	-7	0.28	2	-30	0.07									
48	12	1	0.46	7	-10	0.27	1	-41	0.04						
50	16	8	0.65	10	0	0.41	5	-13	0.20	0					
52	19	13	0.83	14	7	0.61	9	-2	0.39	4	-17	0.18			
54	22	18	1.03	17	12	0.80	12	6	0.56	8	-4	0.38	3	-20	0.14
56	25	22	1.25	20	17	1.00	16	12	0.80	11	5	0.55	7	-5	0.35
58	27	25	1.45	23	21	1.24	18	17	0.97	14	11	0.75	10	4	0.54
59	29	27	1.61	24	23	1.33	20	19	1.11	16	14	0.89	11	8	0.61
60	30	29	1.72	26	25	1.50	21	21	1.21	17	17	0.98	13	11	0.75
61	31	30	1.84	27	27	1.60	22	23	1.31	18	19	1.07	14	14	0.83
62	32	32	1.97	28	29	1.72	24	25	1.47	20	21	1.23	16	16	0.98
63	33	34	2.17	29	30	1.84	25	27	1.59	21	23	1.33	17	19	1.08
64	34	35	2.23	30	32	1.97	26	29	1.71	22	25	1.44	18	21	1.18
65	35	37	2.37	31	34	2.10	27	31	1.83	24	27	1.63	20	24	1.36
66	36	38	2.52	32	35	2.24	29	32	2.03	25	29	1.75	21	26	1.47
67	37	40	2.68	33	37	2.39	30	34	2.17	26	31	1.88	22	28	1.59
68	38	42	2.84	34	39	2.54	31	36	2.32	27	33	2.00	23	29	1.72
69	39	43	3.01	35	40	2.70	32	37	2.47	28	34	2.16	24	31	1.85
70	40	44	3.19	36	42	2.87	33	39	2.63	29	36	2.31	25	33	2.00
71	41	46	3.38	37	43	3.05	33	41	2.72	30	38	2.47	27	35	2.23
72	42	47	3.57	38	45	3.23	34	42	2.89	31	40	2.64	28	37	2.38
73	42	49	3.69	39	46	3.43	35	44	3.07	32	41	2.81	29	38	2.55
74	43	50	3.90	39	48	3.54	36	45	3.26	33	43	2.99	29	40	2.63
75	44	51	4.12	40	49	3.74	37	47	3.46	34	44	3.18	30	42	2.81
76	44	53	4.25	41	51	3.96	38	48	3.67	34	46	3.28	31	43	2.99
77	45	54	4.48	42	52	4.18	39	50	3.89	35	48	3.49	32	45	3.19
78	46	55	4.73	43	53	4.42	39	51	4.01	36	49	3.70	33	46	3.39
79	46	57	4.88	43	55	4.56	40	53	4.24	37	50	3.92	34	48	3.60

TABLE 91.—RELATIVE HUMIDITY, DEW POINTS AND GRAINS OF MOISTURE PER CUBIC FOOT FOR VARIOUS DRY AND WET BULB TEMPERATURES OF THE SLING PSYCHROMETER-BAROMETRIC PRESSURE, 30 INCHES.—(Continued.)

WET BULB DEPRESSION

Dry Bulb Temp.	10			11			12			13		
	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins
80	61	65	6.67	57	63	6.23	54	62	5.90	50	60	5.47
81	61	66	6.88	58	65	6.54	55	63	6.20	51	61	5.75
82	61	67	7.09	58	66	6.74	55	64	6.39	51	62	5.93
83	62	69	7.43	59	67	7.07	56	65	6.71	52	64	6.23
84	62	70	7.66	59	68	7.29	56	66	6.92	52	65	6.43
85	63	71	8.02	60	69	7.64	57	68	7.26	53	66	6.75
86	63	72	8.27	60	70	7.88	57	69	7.48	53	67	6.96
87	64	73	8.66	61	72	8.25	57	70	7.71	54	68	7.17
88	64	74	8.92	61	73	8.50	57	71	7.94	54	69	7.53
89	65	75	9.33	61	74	8.76	58	72	8.33	55	71	7.90
90	65	76	9.61	61	75	9.09	58	73	8.58	55	72	8.13
92	65	79	10.20	62	77	9.73	59	76	9.26	56	74	8.79
94	66	81	10.98	63	79	10.48	60	78	9.98	57	76	9.48
96	66	83	11.63	63	82	11.11	61	80	10.75	58	79	10.22
98	67	85	12.51	64	84	11.95	61	82	11.39	58	81	10.83
100	68	87	13.44	65	86	12.85	62	85	12.26	59	83	11.66
104	69	92	15.27	66	90	14.60	63	89	13.94	60	88	13.28
108	70	96	17.30	67	95	16.56	64	93	15.82	62	92	15.33
112	70	100	19.50	68	99	18.94	65	98	18.11	63	96	17.55
116	71	104	22.05	69	103	21.45	66	102	20.55	64	101	19.95
120	72	108	25.01	69	107	23.95	67	106	23.24	65	105	22.40

Dry Bulb Temp.	14			15			16			17			18		
	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins
80	47	58	5.14	44	56	4.81	41	54	4.48	38	52	4.16	35	50	3.83
81	48	59	5.41	45	57	5.07	42	55	4.74	39	53	4.40	36	51	4.06
82	48	60	5.58	45	59	5.23	42	57	4.88	39	55	4.53	36	52	4.19
83	49	62	5.87	46	60	5.52	43	58	5.01	40	56	4.65	37	54	4.44
84	49	63	6.06	46	61	5.68	43	59	5.31	40	57	4.94	37	55	4.57
85	50	64	6.37	47	62	5.99	44	61	5.60	41	59	5.22	38	57	4.84
86	50	65	6.56	47	64	6.17	44	62	5.78	42	60	5.52	39	58	5.12
87	51	67	6.90	48	65	6.49	46	63	6.22	43	61	5.82	40	59	5.41
88	51	68	7.11	48	66	6.69	46	64	6.41	43	62	5.99	40	61	5.58
89	52	69	7.47	49	67	7.04	47	66	6.75	44	64	6.32	41	62	5.89
90	52	70	7.69	49	69	7.25	47	67	6.95	44	65	6.51	41	63	6.06
92	53	73	8.32	50	71	7.84	48	69	7.53	45	68	7.06	42	66	6.59
94	54	75	8.98	51	73	8.48	49	72	8.15	46	70	7.65	43	68	7.15
96	55	77	9.69	52	76	9.17	50	74	8.81	47	72	8.28	44	71	7.76
98	56	79	10.46	53	78	9.90	50	76	9.34	48	75	8.96	45	73	8.40
100	56	82	11.07	54	80	10.67	51	79	10.08	49	77	9.69	46	76	9.09
104	58	86	12.83	55	85	12.17	53	83	11.73	50	82	11.06	48	80	10.62
108	59	91	14.59	57	89	14.09	54	88	13.35	52	86	12.85	49	85	12.11
112	60	95	16.72	58	94	16.16	55	92	15.39	53	91	14.84	51	90	14.28
116	61	99	19.05	59	98	18.45	57	97	17.85	54	95	16.87	52	94	16.21
120	62	104	21.46	60	102	20.75	58	101	20.07	55	100	19.05	53	98	18.37

TABLE 91.—RELATIVE HUMIDITY, DEW POINTS AND GRAINS OF MOISTURE PER CUBIC FOOT FOR VARIOUS DRY AND WET BULB TEMPERATURES OF THE SLING PSYCHROMETER-BAROMETRIC PRESSURE, 30 INCHES.—(Continued.)

	WET BULB DEPRESSION																										
	19			20			21			22			23			24			25			26			27		
Dry Bulb Temp.	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins
56	2	-25	0.10																								
58	6	-6	0.32	1	-30	0.05																					
60	9	4	0.52	5	-8	0.29	1	-36	0.06																		
62	12	11	0.74	8	3	0.49	4	-9	0.25	1	-45	0.06															
64	15	17	0.98	11	11	0.72	7	3	0.46	4	-10	0.26	0														
66	17	22	1.19	14	17	0.98	10	11	0.70	7	2	0.49	3	-11	0.21	0											
68	20	26	1.50	16	22	1.20	13	17	0.97	10	11	0.75	6	2	0.45	3	-11	0.22									
70	22	30	1.76	19	26	1.52	15	22	1.20	12	17	0.96	9	11	0.72	6	2	0.48	3	-11	0.24						
72	24	33	2.04	21	30	1.79	18	26	1.53	15	22	1.28	12	17	1.02	9	11	0.77	6	3	0.51	3	-11	0.26			
74	26	37	2.36	23	34	2.09	20	30	1.81	17	27	1.54	14	23	1.27	11	18	1.00	8	12	0.73	5	3	0.45	3	-10	0.27
75	27	39	2.53	24	36	2.25	21	32	1.96	18	29	1.68	15	25	1.40	12	21	1.12	9	15	0.84	7	8	0.66	4	-2	0.37
76	28	41	2.70	25	38	2.41	22	34	2.12	19	31	1.84	16	27	1.55	13	23	1.26	11	18	1.06	8	12	0.77	5	4	0.48
77	29	42	2.89	26	39	2.59	23	36	2.29	20	33	1.99	17	29	1.69	14	26	1.39	12	21	1.20	9	16	0.90	6	9	0.60
78	30	44	3.08	27	41	2.78	24	38	2.47	21	35	2.16	18	31	1.85	16	28	1.64	13	24	1.34	10	19	1.03	8	13	0.82
79	31	46	3.29	28	43	2.97	25	40	2.65	22	37	2.33	19	34	2.01	17	30	1.80	14	26	1.48	11	22	1.17	9	16	0.95
80	32	47	3.50	29	44	3.17	26	42	2.74	23	39	2.52	20	36	2.19	18	32	1.97	15	28	1.64	12	24	1.31	10	20	1.09
81	33	49	3.72	30	46	3.38	27	43	3.04	24	41	2.71	21	38	2.37	19	34	2.14	16	31	1.80	13	27	1.47	11	22	1.24
82	33	50	3.84	30	48	3.49	28	45	3.26	25	42	2.91	22	39	2.56	20	36	2.33	17	33	1.98	14	29	1.63	12	25	1.40
83	34	52	4.08	31	49	3.72	29	47	3.48	26	44	3.12	23	41	2.76	21	38	2.52	18	35	2.16	15	31	1.80	13	27	1.56
84	35	53	4.33	32	51	3.95	29	48	3.58	26	46	3.31	24	43	2.97	21	40	2.59	19	37	2.35	16	34	1.98	14	30	1.73
85	36	54	4.59	33	52	4.20	30	50	3.82	27	48	3.44	25	45	3.18	22	42	2.80	20	39	2.55	17	36	2.17	15	32	1.91
86	36	56	4.73	33	54	4.33	31	52	4.07	28	49	3.68	26	47	3.41	23	44	3.02	21	41	2.76	18	38	2.36	16	34	2.10
87	37	57	5.00	34	55	4.60	32	53	4.33	29	51	3.92	26	48	3.52	24	46	3.25	22	43	2.98	19	40	2.57	17	36	2.30
88	37	59	5.16	35	57	4.88	32	55	4.46	30	52	4.18	27	50	3.76	25	47	3.48	22	45	3.07	20	42	2.79	18	38	2.51
89	38	60	5.46	36	58	5.17	33	56	4.74	31	54	4.45	28	51	4.02	26	49	3.73	23	46	3.30	21	44	3.02	19	41	2.73
90	39	61	5.77	36	59	5.33	34	57	5.03	31	55	4.59	29	53	4.29	26	51	3.85	24	48	3.55	22	45	3.25	19	43	2.81
92	40	64	6.28	37	62	5.81	35	60	5.49	32	58	5.02	30	56	4.71	28	54	4.39	25	51	3.92	23	49	3.61	21	46	3.30
94	41	67	6.82	38	65	6.32	36	63	5.99	33	61	5.49	31	59	5.16	29	57	4.82	27	55	4.49	24	52	3.99	22	50	3.66
96	42	69	7.40	39	67	6.87	37	66	6.52	35	64	6.17	32	62	5.64	30	60	5.29	28	58	4.94	26	55	4.58	24	53	4.23
98	43	72	8.03	40	70	7.47	38	68	7.10	36	66	6.72	34	64	6.35	32	63	5.98	29	61	5.41	27	58	5.04	25	56	4.67
100	44	74	8.70	41	72	8.10	39	71	7.71	37	69	7.31	35	67	6.92	33	65	6.52	30	63	5.93	28	61	5.54	26	59	5.14
104	46	79	10.18	43	77	9.51	41	76	9.07	39	74	8.63	37	72	8.19	35	71	7.75	33	69	7.30	31	67	6.86	29	65	6.42
108	47	84	11.62	45	82	11.12	43	81	10.63	41	79	10.14	39	77	9.64	37	76	9.15	35	74	8.65	33	72	8.16	31	71	7.66
112	49	88	13.70	47	87	13.12	44	85	12.26	42	84	11.68	40	82	11.10	38	81	10.54	36	79	9.98	35	78	9.42	33	76	8.86
116	50	93	15.62	48	91	14.98	46	90	14.33	44	88	13.69	42	87	13.04	40	86	12.40	38	84	11.78	36	83	11.16	34	81	10.56
120	51	97	17.69	49	96	16.96	47	94	16.28	45	93	15.60	43	92	14.92	41	90	14.24	40	89	13.90	38	87	13.20	36	86	12.49

Table 91.—Relative Humidity, Dew Points and Grains of Moisture per Cubic Foot for Various Dry and Wet Bulb Temperatures of the Sling Psychrometer-Barometric Pressure, 30 Inches.—(Concluded.)

	Wet Bulb Depression																							
	28			29			30			31			32			33			34			35		
Dry Bulb Temp.	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins	%	Dew Point	No. of Gr'ins
75	1	−23	0.09																					
76	3	10	0.29																					
77	4	−2	0.40	1	−21	0.10																		
78	5	5	0.51	3	−9	0.31																		
79	6	10	0.64	4	−1	0.42	1	−20	0.11															
80	7	13	0.77	5	6	0.55	3	−7	0.33															
81	9	17	1.02	6	10	0.68	4	0	0.45	1	−18	0.11												
82	10	20	1.16	7	14	0.81	5	7	0.58	2	−6	0.23	0											
83	11	23	1.32	8	18	0.96	6	11	0.72	3	2	0.36	2	−15	0.24									
84	12	26	1.48	9	21	1.11	7	15	0.87	5	8	0.62	3	−4	0.37	0								
85	13	28	1.66	10	24	1.27	8	19	1.02	6	12	0.77	4	3	0.51	2	−12	0.26						
86	14	31	1.84	11	27	1.44	9	22	1.18	7	16	0.92	5	9	0.66	3	−2	0.39	1	−27	0.13			
87	15	33	2.03	12	29	1.62	10	25	1.35	8	20	1.08	6	13	0.81	4	5	0.54	2	−10	0.27			
88	15	35	2.09	13	31	1.81	11	27	1.53	9	23	1.26	7	17	0.98	5	10	0.70	3	0	0.42	1	−22	0.14
89	16	37	2.30	14	34	2.01	12	30	1.72	10	26	1.44	8	21	1.15	6	15	0.86	4	6	0.57	2	−7	0.29
90	17	39	2.52	15	36	2.22	13	32	1.92	11	28	1.63	9	24	1.33	7	19	1.04	5	11	0.74	3	1	0.44
92	19	43	2.98	17	40	2.67	15	37	2.35	13	33	2.04	11	29	1.73	9	25	1.41	7	20	1.10	5	13	0.78
94	20	47	3.33	18	44	2.99	16	41	2.66	14	38	2.33	12	34	2.00	10	30	1.66	9	26	1.50	7	21	1.16
96	22	51	3.88	20	48	3.53	18	45	3.17	16	42	2.82	14	39	2.47	12	35	2.12	10	31	1.76	8	27	1.41
98	23	54	4.29	21	52	3.92	19	49	3.55	17	46	3.17	15	43	2.80	14	40	2.61	12	36	2.24	10	32	1.87
100	24	57	4.74	22	55	4.35	21	52	4.15	19	50	3.76	17	47	3.36	15	44	2.97	13	41	2.57	12	37	2.37
104	27	63	5.98	25	61	5.53	23	59	5.09	21	57	4.65	20	54	4.43	18	52	3.98	16	49	3.54	14	46	3.10
108	29	69	7.17	27	67	6.67	25	65	6.18	24	63	5.93	22	61	5.44	20	59	4.94	19	56	4.70	17	54	4.20
112	31	74	8.58	29	72	8.03	27	71	7.49	26	69	7.22	24	67	6.68	23	65	6.41	21	63	5.87	19	61	5.33
116	33	79	10.25	31	78	9.61	29	76	8.99	28	74	8.68	26	73	8.06	25	71	7.78	23	69	7.17	22	67	6.86
120	34	84	11.79	33	83	11.44	31	81	10.73	29	80	10.04	28	78	9.70	26	76	9.02	25	75	8.68	23	73	7.99

creased depression of the reading of the wet bulb thermometer below the reading of the temperature of the dry bulb.

The relation between the pressure of the dry air and the water vapor in the mixture is determined by assuming that Dalton's law will hold good for vapors at low pressures, and that the actual pressure of the vapor in the mixture and the pressure of the saturated vapor at the same temperature have a ratio which is equivalent to the relative humidity. This may be expressed in a formula as follows:

$$P_v = h \times P_{sv}$$

where P_v = actual pressure of the vapor in mixture at temperature of the mixture
h = relative humidity of the mixture
P_{sv} = pressure of the saturated vapor at the temperature of the mixture

The relation between the total pressure and the partial pressure of the air and the vapor may be expressed as follows:

$$P_m = P_a + P_v \text{ or}$$
$$P_a = P_m - P_v$$

The relation between the pressure, volume and temperature of the mixture may be expressed in the following equation:

$$(P_m - P_v) \times V_a = 0.37\, T_v$$

The weight of the dry air per cubic foot may be found as follows:

$$w_a = \frac{1}{V_a}$$

The weight of the water vapor in pounds per cubic foot of the mixture may be determined by the following formula:

$$w_v = h \times w_{sv}$$

where w_v = weight of water vapor in a cubic foot in lbs.
h = relativ humidity
w_{sv} = weight of the saturated water vapor per cu. ft. at the temperature of the mixture

The total weight of the mixture per cubic foot may be obtained as follows:

$$w_m = w_a + w_v$$

The total weight of water vapor contained in a pound of dry air may be found as follows:

$$w = V_a \times w_v$$

where w = weight of water vapor per lb. of dry air

In general, to obtain a thorough understanding of the mechanical cooling of air, one must appreciate the relation between the wet and dry bulb temperatures, the dew point, and the relative humidity. Air that is saturated has a dew point and dry bulb and wet bulb temperature readings which are identical.

It is evident that if air that is saturated is cooled, the volume will be contracted and some of the moisture will be condensed. The magnitude of the cooling effect required will be determined by the initial and final temperatures of the saturated air. When the air is partially saturated, the cooling process is somewhat different. As the temperature of the air is reduced, or as the heat is removed from the air, the dry bulb temperature falls and the wet bulb temperature falls until they finally reach the dew point temperature, at which the air is completely saturated. The cooling to be effected in this case, therefore, amounts to the sensible heat of the air and the latent heat of condensation of the moisture.

In some cases, it is desirable to pass the air through a fine spray of recirculating water. In the case that the air is partially saturated, the air will absorb moisture; the dry bulb temperature will begin to fall and will continue to fall until it reaches the wet bulb temperature. The dew point, at the same time, will have risen to the wet bulb temperature and the air will be completely saturated. It is evident that heat is interchanged between the water and the air, and that very little heat will be taken from any other source. The sensible heat evolved from lowering the temperature of the air appears again in the form of latent heat of vaporization of the water which has been evaporated to saturate the air. This is known as the adiabatic saturation of air. From this it will be observed that when the air is not adiabatically saturated with moisture, the cooling will be that required to cool the air from the initial dry bulb temperature to the dew point temperature, together with the latent heat of condensation of the moisture; that when the air is adiabatically saturated with moisture, the heat required to be removed from the air will depend upon the heat content of the saturated air at the wet bulb temperature and the heat content of the saturated air at the lowest temperature.

The magnitude of the heat content may be assumed to consist of the sensible heat of the air and the latent heat of the vaporization of the moisture in the air, neglecting the heat of the liquid of the water, which is a very small quantity. This may be expressed as follows:

$$H_m = C_{pa} \times t_m + r_v \times w$$

where H_m = heat content of the mixture Btu. per lb. above 0° F.

t_m = temperature of the mixture degrees F.

r_v = latent heat of vaporization of the water vapor at the temperature of mixture

Or in the case that the air is adiabatically saturated with moisture, the heat content will be found to be equal to the sensible heat of the air at the temperature of the wet bulb and the latent heat of vaporization of the water vapor at the temperature of the wet bulb. This may be expressed as follows:

$$H_{wb} = C_{pa} \times t_{wb} + r_v \times w$$

where H_{wb} = heat content of the air at wet bulb temperature in Btu. per lb. above 0° F.

t_{wb} = temperature wet bulb thermometer in degrees F.

Example 1.—Atmospheric air is at the normal pressure of 14.7 pounds per square inch and has a dry bulb temperature of 90° and a wet bulb temperature of 85°. It is desired to determine the relative humidity, the dew point, the partial air and water vapor pressures, the volume of a pound of the mixture, the relative weights of dry air and water vapor per cubic foot of the mixture, and the heat content of the mixture in Btu. per pound above 0° F.

By referring to Table 91 it will be noted that the relative humidity corresponding to the dry bulb temperature of 90° and the wet bulb temperature of 85° is 81 per cent, and at the same point it will be noted that the dew point temperature is 83°. Also, from Table 89, it will be noted that the saturated water vapor pressure corresponding to 90° is 0.696 pounds per square inch. The various quantities may be calculated as shown in the following paragraph:

The absolute pressure of the saturated water vapor in the mixture in pounds per square inch may be found as follows:

$$P_v = 0.81 \times 0.696 = 0.564 \text{ lbs.}$$

The absolute pressure of dry air in the mixture, in pounds per square inch, is found as follows:

$$P_a = 14.700 - 0.564 = 14.136 \text{ lbs.}$$

The volume of the dry air in cubic feet is found as follows:

$$V_a = \frac{0.37\ T_a}{P_m - P_{sv}} = \frac{0.37\ T_a}{P_a}$$

$$= \frac{0.37 \times 549.6}{14.136} = 14.35$$

The weight of a cubic foot in pounds is found as follows:

$$w_a = \frac{1}{14.35} = 0.0698$$

The weight of vapor in one cubic foot of the mixture is found as follows:

$$w_v = 0.81 \times \frac{1}{469.3} = 0.00173$$

The weight of the mixture in pounds per cubic foot is found as follows:

$$w_m = 0.0698 + 0.00173 = 0.07153$$

The weight of water vapor in one pound of dry air is found as follows:

$$w = 0.00173 \times 14.35 = 0.0248$$

The mean specific heat of dry air at a constant pressure between the temperatures of t_1 and t_2 is found as follows:

$$\begin{aligned} C_{pa} &= 0.2411 + 0.0000045\ (0 + 90) \\ &= 0.2415 \end{aligned}$$

The heat content of the mixture above 0° F. is found as follows:

$$\begin{aligned} H_m &= 0.2415 \times 90 + 0.0248 \times 1041.2 \\ &= 47.6 \end{aligned}$$

The heat content of the saturated air at the temperature of the wet bulb thermometer may be determined as follows:

$$H_{wb} = C_{pa} \times t_{wb} + r_v \times w$$

The volume of one pound of dry air is found as follows:

$$\begin{aligned} V_a &= \frac{0.37\ T_a}{P_m - P_{av}} = \frac{0.37 \times (85 + 459.6)}{14.7 - 0.594} \\ &= 14.3 \end{aligned}$$

Water vapor per pound of air is calculated as follows:

$$w = 14.3 \times \frac{1}{545.9} = 0.0262$$

Then the heat above 0° F. is found to be:

$$\begin{aligned} H_{wb} &= 0.2415 \times 85 + 1044 \times 0.0262 \\ &= 48.0 \end{aligned}$$

Psychrometric Properties of Air.—The various psychrometric properties of dry, saturated, and partially saturated air have been thoroughly investigated by W. H. Carrier. The result of his research is shown by Fig. 172 in cover pocket, which indicates graphically the relationship of the dry and wet bulb temperatures, the dew point temperatures, percentages of humidity, specific volumes, and heat contents. Mr. Carrier presented his findings before the A. S. M. E. at the 1911 annual meeting.

Referring to Fig. 172, it will be noted that the dry bulb temperatures are represented by vertical lines with values indicated on lower edge of chart. Wet bulb temperatures are represented by oblique straight lines with values indicated on the curved line marked "Wet Bulb Temperature." Dew point temperatures are represented by horizontal lines and their values indicated on the curved line, *A*, marked "Wet Bulb Temperature." Percentages of relative humidity are represented by converging curved lines with values indicated between the oblique straight lines for 63° and 64° F. wet bulb temperature. Any two of the properties may be found if two others are known. First, find the point of intersection of the lines representing the given properties, and then follow through this point, the lines representing the unknown properties and the values of the latter can be read from their respective scales.

Total heat of air is a function of its wet bulb temperature, as described and proved by Mr. Carrier in his paper, "Rational Psychrometric Formulae." In the chart, it is taken for the sake of convenience as above 0° F. The total heat in Btu. is represented by horizontal lines with values indicated at the left by the scale *D*. "Total Heat Above Zero Degrees Contained in One Pound of Dry Air Saturated with Moisture."

To find the total heat, the wet bulb temperature must first be determined. Then follow the vertical line through the wet bulb temperature reading at 100 per cent relative humidity, to the "Total Heat," curve *D*. The horizontal line through this point of intersection represents the total heat above 0° F., and the value is indicated on the scale *D* at the left.

To find the resulting relative humidity when heating or cooling air without changing its dew point, the following procedure is necessary: Find the dew point line for the given conditions, which is the horizontal line through the point of intersection of the (vertical) dry bulb line with the (oblique) wet bulb line. Follow this line horizontally to the new dry bulb temperature line and read the percentage of humidity for this point of intersection.

Vapor pressure in inches of mercury depends upon the dew point and is represented by horizontal lines with values indicated by the scale *E*, "Vapor Pressure in Inches of Mercury." To determine the vapor pressure read horizontally from intersection of (vertical) dry bulb line with (oblique) wet bulb line to the wet bulb temperature, curve *A*. Then read vertically to the vapor pressure curve *E*, and then to the vapor pressure scale *E*, at the left. The reading here is the vapor pressure. It should be noted when getting the reading for saturated air that the dew point, wet and dry bulb temperatures are identical.

Absolute humidity (grains of moisture per cubic foot), may be found as follows: Absolute humidity is a function of the dew point and may be expressed either in grains per pound of dry air or in grains per cubic foot. The grains of moisture per pound of dry air are represented by horizontal lines with values indicated, on the scale *A*, "Grains of Moisture per Pound of Dry Air."

To find the grains of moisture per cubic foot for air under given conditions, first determine the grains per pound of dry air by finding the intersection of the (vertical) dry bulb line and the (oblique) wet bulb line and reading from there horizontally to scale *A*, "Grains of Moisture per Pound of Dry Air" at the left. The volume in cubic feet per pound of dry air is represented by horizontal lines with values indicated by scale *B*, "Cubic Feet per Pound." Find the points of intersection of the (vertical) dry bulb temperature line with the curves *B*, "Volume in Cubic Feet of One Pound of Dry Air Saturated with Moisture" and "Volume in Cubic Feet of One Pound of Dry Air." Follow these horizontally to the left and read the values on scale *B*. The difference in these two readings of volumes is the extra volume due to moisture at 100 per cent relative humidity.

To obtain direct the volume of the moisture per pound of air under the given conditions, find the volume of water vapor for the given per cent of humidity and add it to the volume of dry air.

Then grains of moisture per cubic foot, or in other words, the absolute humidity is found as follows:

$$\frac{\text{Grains per pound of dry air}}{\text{Cubic feet per pound of mixture}}$$

The following examples will indicate the usefulness of the Carrier Psychrometric Chart, Fig. 172.

Example 1. Given: Dry bulb temperature, 70° F.; wet bulb temperature, 60° F. Find the percentage of relative humidity and the dew point.

Locate point of intersection of vertical line representing 70° dry bulb temperature and the oblique line representing 60° wet bulb temperature. Reading from this point, the percentage of relative humidity is 56 and the dew point 53.4° F.

Example 2. Given: Dry bulb temperature, 80°; relative humidity, 59 per cent. Find the dew point and wet bulb temperature.

Locate the point of intersection of the vertical line representing 80° dry bulb temperature and the curved line representing 59 per cent relative humidity. Reading from this point, the dew point is 64.8° and the wet bulb temperature 69.5°.

Example 3. Given: Dry bulb temperature, 75°; dew point temperature, 55°. Find percentage of relative humidity and wet bulb temperature.

Locate the point of intersection of the vertical line representing 75° dry bulb temperature and the horizontal line representing 55° (wet bulb temperature scale). Reading from this point, the relative humidity is 49.8 per cent and the wet bulb temperature 62.3°.

Example 4. Given: Relative humidity, 50 per cent; wet bulb temperature, 60°. Find dry bulb temperature and dew point.

Locate the point of intersection of the curved line representing 50 per cent relative humidity and the oblique line representing 60° wet bulb temperature. Reading from this point, the dry bulb temperature is 71.7° and the dew point 52°.

Example 5. Given: Wet bulb temperature, 55°; dew point, 50°. Find dry bulb temperature and relative humidity.

Locate the point of intersection of the oblique line representing 55° wet bulb temperature and the horizontal line representing dew point of 50°. Reading from this point, the dry bulb temperature is 61.5° and the relative humidity 68 per cent.

Example 6. Given: Relative humidity, 40 per cent; dew point, 40°. Find dry bulb temperature and wet bulb temperature.

Locate the point of intersection of the curved line representing 40 per cent relative humidity and the horizontal line representing 40° dew point. Reading from this point, the dry bulb temperature is 65° and the wet bulb temperature 52°.

Example 7. Given: Dry bulb temperature, 70°; relative humidity, 60 per cent. Find the total heat.

The wet bulb temperature determined by the method used in Example 2, is 61° F. Following the vertical line through the point representing 61° wet bulb temperature and 100 per cent relative humidity, to its intersection with curve *D* and then reading horizontally to scale *D* on the left, the total heat above zero degrees in one pound of given air is 26.7 Btu.

Example 8. Given: Dry bulb temperature, 70°; wet bulb temperature, 60°. Find the resulting relative humidity when air is heated to 80° (dry bulb temperature).

The point of intersection of (vertical) 70° dry bulb and (oblique) 60° wet bulb line indicates an existing humidity of 56 per cent. Following the point of this indication horizontally to intersection with the (vertical) 80° dry bulb line the relative humidity reading on the curve is 40 per cent. The heat content at dry bulb temperature of 70° and wet bulb temperature of 60° is noted to be 26.0 Btu. The heat content at 80° and 40 per cent relative humidity is observed to be 28.4. The

difference, 28.4 — 26.0 = 2.4 Btu., the heat required to heat or cool the air under these conditions.

Example 9. Given: Dry bulb temperature, 70°; wet bulb temperature 60°. Find the vapor pressure.

Read vertically on the 70° dry bulb line to the point of intersection with the (oblique) 60° wet bulb line, and then horizontally to the wet bulb temperature curve *A*, locating the dew point, then read vertically from the dew point 53.5°, to the vapor pressure curve *E*, and then to vertical scale *E*, "Vapor Pressure in Inches of Mercury." The vapor pressure for this point reads 0.406 in.

Example 10. Given: Dry bulb temperature, 70°; wet bulb temperature, 60°. Find the absolute humidity (grains of moisture per cubic foot).

Reading vertically on the 70° dry bulb line and obliquely on the 60° wet bulb line to the curve marked "Wet bulb temperature," and from there directly horizontally to scale *A*, the number of grains of moisture per pound of dry air is 61.

Then locate the points of intersection of the vertical dry bulb line for 70° with the two curves *B*, and follow these horizontally to scale *B* at the left, this gives the following volumes:

13.7 cu. ft.—Volume of 1 lb. of dry air saturated with moisture
13.38 cu. ft.—Volume of 1 lb. of dry air

0.32 cu. ft.—Extra volume due to moisture at 100 per cent humidity

In instances where the air is saturated or dry, its volume may be found directly by locating the point of intersection of the (vertical) dry bulb line with the corresponding curve *B* and referring to the scale *B*.

The percentage of relative humidity for air under the given conditions is indicated where the 70° dry and 60° wet bulb lines intersect, which by interpolating between the 60 per cent and 50 per cent relative humidity curves is 56 per cent. The products 0.56 × 0.32 gives the cubic ft. volume of moisture in the given air as 0.1792. The total volume is found as follows:

13.38 cu. ft.—Volume of dry air
0.1792 cu. ft.—Volume of moisture

13.5592 cu. ft.—Volume of 1 lb. of the given mixture

The quotient, 61 ÷ 13.5592 = 4.50 grains of moisture per cubic foot.

Refrigeration Requirements.—From the foregoing, it will be noted that atmospheric air is composed primarily of dry air and water vapor. Therefore, in considering the amount of cooling which is required to

change the temperature of the air, it is evident that consideration must be given to the sensible heat of the air and the latent heat of the water vapor. In addition to this, it is evident that a certain amount of refrigeration is necessary for cooling the water that is condensed out of the air as it is cooled. In general, this factor is quite small and may be neglected. Also, if the moisture that is condensed out of the air is allowed to freeze on the refrigerating surfaces, it is evident that allowance must be made for the latent heat of fusion of the ice.

The principal refrigeration requirements depend upon the sensible heat of the air and the latent heat of fusion of the moisture. The amount of cooling that must be allowed for these two effects may be determined by noting the difference between the heat contents of the air before and after cooling. The method of calculation of these heat contents has been previously indicated, and such heat contents may be rapidly estimated by means of Table 91 and Fig. 172. When the condition of the air is known before cooling, the wet bulb temperature may be determined by means of Table 91 and Fig. 172. After this wet bulb temperature has been determined, and remembering that the heat content depends upon the saturated air at the temperature of the wet bulb thermometer, this heat content may be taken directly from Fig. 172 or it may be calculated as previously outlined.

In a similar manner, it is possible to determine the heat content of the air after it has been cooled. It should be noted that these heat contents, as indicated on Fig. 172, give only the heats which are necessary for the sensible heat of the air and the latent heat of the moisture. If it is desired to give consideration to the heat of the liquid and the latent heat of fusion of the ice, the magnitudes of these heats may be estimated as follows: The amount of water vapor present in the air before cooling is readily determined by calculations as previously outlined or may be read directly from the chart. In a similar manner, the amount of water vapor present in the air after it has been cooled may be calculated or taken from Fig. 172. By noting the difference between these two vapor contents, the amount that is condensed out is determined. After determining the exact amount of water vapor which is condensed, the heat to be removed may be taken to be proportional to one-half of the temperature range of the air.

The heat to be removed to freeze the water which is condensed out of the air may be allowed for in direct proportion to the weight of the water condensed out and the latent heat of fusion of the ice. It should be noted that any condition of relative humidity may be secured, or any temperature of air may be maintained by the use of mechanical refrigeration. The humidity of the air may be controlled by cooling the air to a temperature which is sufficiently below the desired tem-

perature, that the moisture content at the lower temperature will give the proper humidity when the air is heated to the desired temperature.

The process simply consists of cooling the air to a low temperature to reduce the moisture content, after which the air is heated at constant pressure to the desired temperature before it is introduced into the room or process under consideration. It is evident that the amount of cooling and the amount of heat required in the latter case may be calculated by the methods which have been previously indicated.

Example 1. In the previous example, the atmospheric air had a dry bulb temperature of 90° and a wet bulb temperature of 85°, and was under a normal atmospheric pressure of 14.7 lb. per sq. in. It is desired to cool and condition this air to a temperature of 65° and a relative humidity of 61 per cent. It is desired to determine how low the air may be cooled, so that when it is heated to 65° it has a relative humidity of 61 per cent; to determine the amount of initial cooling effect; to determine the amount of heat to be added to bring the air up to the temperature of 65°.

Since the heat content of the air at a dry bulb temperature of 90° and a wet bulb temperature of 85° depends upon the temperature of the wet bulb thermometer, the heat content of this condition may be taken from Fig. 172, or it may be calculated, as previously indicated. If taken from the chart, it will be observed that the heat content of dry saturated air at 85° F. is 47.7 Btu. per lb. If it is desired to have the air at a temperature of 65° and a relative humidity of 61 per cent, it is evident that the air must be cooled to the dew point temperature so that when it is heated up to 65° the moisture content present will give the air a humidity of 61 per cent.

From Fig. 172, therefore, it will be observed that the dew point corresponding to the temperature of 65° and a relative humidity of 61 per cent is 51° F. Thus, the air in this problem must be cooled from a temperature of 90° to a temperature of 51°. The heat content at 51° and 100 per cent relative humidity may be obtained by calculation or by reading from the chart. The content of dry saturated air at 51° is 20.8 Btu. per lb. The difference in the heat content before and after cooling will give the cooling effect required, which in this case is equal to $47.7 - 20.8 = 26.9$. In order to determine what heat must be added to this air to heat it from 51° to 65°, it is only necessary to note the heat contents at these temperatures.

From Table 91, it will be observed that the depression of the wet bulb thermometer for a temperature of 65° and a relative humidity of 61 per cent is equal to 8°, since the wet bulb thermometer reading is 57°. The heat content of dry saturated air at 57° and a 100 per cent relative humidity may be taken from Fig. 172 and is equal to 24.2. The

heat required to reheat the air from 51° to 65° would be equal to $24.2 - 20.8 = 3.4$ Btu. per lb.

The heat required to cool the moisture which is condensed out of the air may be approximately obtained as follows: From Fig. 172 it will be observed that the moisture content of air at 90° dry bulb and 85° wet bulb temperature is equal to 175 grains per lb. of dry air. At 51° the water vapor to saturate one pound of air may be obtained from Fig. 172 and will be observed to be 56 grains. The heat required to cool the moisture which is condensed out would be found as follows:

$$\left(\frac{175-56}{7000}\right) \times \left(\frac{85-51}{2}\right) = 0.290$$

The total cooling in this case would be $27.3 + 0.290 = 27.59$ Btu. per lb.

Weather Conditions.—The refrigeration required for the mechanical cooling of air depends, to a certain extent, upon the general weather conditions and the location of the plant. It is evident that in estimating the amount of refrigeration requirement the average maximum weather conditions for the plant in question should be taken into consideration. The refrigerating plant must, at all times, be able to produce the desired result, so that, in general, it is advisable to take into consideration the average maximum weather conditions when designing the plant.

The average extreme weather conditions for cities located in the United States and some foreign countries are shown by Table 92. This table gives the dry bulb temperature, the wet bulb temperature, the relative humidity, and the direction and velocity of the prevailing winds. The values shown in the table are averages for the month of July and have been based upon government data from the United States Weather Bureau. It will be noted that the weather conditions or the state of the atmosphere varies greatly with the locality. In addition to the variations shown, it is evident that the conditions of the atmosphere will vary from day to day so that when an air cooling plant is being considered, it is well to use temperatures given in Table 92 or temperatures which are somewhat higher. If the refrigerating plant is designed under these considerations, it will in general give satisfaction and service.

Types of Air Cooling Apparatus.—The apparatus used for the cooling of air may be divided into two general classes, according to the principles of operation. In the first class are those which cool the air by means of cold surfaces, and in the second class are those that cool the air by means of cold liquids. The principal results of this

TABLE 92.—EXTREME WEATHER CONDITIONS.
(JULY AVERAGES.)
The Cooling Tower Company

State	City	Dry bulb in Deg. Fahr.	Relative Humidity in Per Cent.	Wet bulb in Deg. Fahr.	Wind Miles per Hour	Direction of Wind
Alabama	Montgomery	81.0	70.0	73.5	5.0	S. W.
	Mobile	80.5	79.0	75.5	6.0	S.
	Birmingham	79.8	73.0	73.3	5.0	S. W.
Arizona	Yuma	91.0	36.0	71.0	....	W. & S.
	Phoenix	90.5	31.0	68.5	....	E.
Arkansas	Little Rock	80.0	72.0	73.0	5.0	S. & S. W.
	Ft. Smith	80.5	68.0	72.5	5.0	E.
California	San Fran.	57.3	82.0	54.3	14.0	W.
	Sacramento	72.4	51.0	60.6	9.7	S.
	Los Angeles	67.4	72.0	61.6	4.5	W.
	Red Bluff	82.1	30.0	62.0	5.7	S. E.
	San Diego	66.9	78.0	62.4	5.4	S. W.
Colorado	Denver	71.8	45.0	57.8	7.5	S.
	Colo. Springs	67.9	51.0	56.4	....	....
	Pueblo	72.6	44.0	58.0	7.0	....
Connecticut	New Haven	71.9	73.0	65.9	8.0	S.
Dist. Col.	Washington	76.8	69.0	69.3	5.3	S.
Florida	Jacksonville	80.9	75.0	75.0	8.0	S. W.
	Key West	83.7	73.0	76.7	8.0	E.
	Tampa	79.9	79.0	75.0	6.0	E. & N. E.
	Pensacola	81.3	77.0	75.8	8.0	S. W.
Georgia	Augusta	80.5	74.0	74.0	5.0	S. & S. E.
	Atlanta	77.6	67.0	69.6	8.6	N. W.
	Savannah	80.5	79.0	75.5	6.4	S. W.
Illinois	Cairo	78.6	75.0	72.4	6.2	S.
	Chicago	72.3	71.0	65.8	15.1	S. W.
	Springfield	76.1	64.0	68.0	6.6	S. W.
Indiana	Indianapolis	76.4	65.0	68.0	8.2	S. W.
Iowa	Davenport	75.4	65.0	67.2	7.4	S. W.
	Des Moines	75.0	66.0	67.0	7.1	S. W.
	Dubuque	74.7	66.0	66.7	5.5	N. W.
	Keokuk	77.0	67.0	69.0	6.9	S.
Kansas	Concordia	77.7	58.0	67.2	....	S.
	Dodge City	77.7	58.0	66.7	....	S. E.
	Wichita	78.3	65.0	69.5	....	S.
Kentucky	Lexington	76.0	64.0	68.0	8.0	S. W.
	Louisville	78.6	64.0	69.6	6.1	S. W.
Louisiana	New Orleans	81.3	76.0	75.3	6.5	S. W.
	Shreveport	82.1	72.0	75.0	5.0	S. & S. E.
Maine	Eastport	59.8	81.0	56.3	....	S. & S. W.
	Portland	68.0	71.0	62.0	....	S.
Maryland	Baltimore	77.3	70.0	69.6	6.6	S. W.
Mass.	Boston	71.3	70.0	64.8	9.3	S. W.
	Nantucket	67.5	85.0	64.5	....	S. W.
Michigan	Alpena	65.8	78.0	60.3	8.0	S. E. & W.
	Detroit	72.1	69.0	65.1	9.0	S. W.
	Grand Haven	69.7	70.0	63.2	9.0	S. W.
	Escanaba	66.7	73.0	61.2	8.0	S.
	Marquette	64.9	70.0	59.0	9.0	N. W.
	Port Huron	69.0	72.0	63.0	9.0	N. E.
	Grand Rapids	72.6	68.0	65.4	9.0	S. W.

TABLE 92.—EXTREME WEATHER CONDITIONS.
(JULY AVERAGES.)—(Continued.)

State	City	Dry bulb in Deg. Fahr.	Relative Humidity in Per Cent.	Wet bulb in Deg. Fahr.	Wind Miles per Hour	Direction of Wind
Miss.	Vickburg	80.4	75.0	74.4	6.0	S. W.
Minn.	St. Paul	72.1	67.0	64.5	7.2	N.
	Duluth	66.0	71.0	60.0	11.0	N. E.
Missouri	Kansas City	76.9	67.0	68.9	7.5	S.
	St. Louis	79.1	66.0	70.6	8.2	S. W.
	Springfield	75.7	70.0	68.5	7.7	S.
Montana	Havre	68.5	50.0	57.0	....	S. W.
	Helena	66.9	43.0	53.4	....	W.
	Miles City	73.0	55.0	62.0	....	W. & N. W.
Nebraska	North Platte	73.9	61.0	64.9	9.0	S. E.
	Omaha	76.5	59.0	66.5	7.0	S.
	Valentine	73.1	57.0	62.5	10.0	S.
New Jersey	Atlantic City	72.5	84.0	69.0	8.0	S. W.
New York	Albany	72.0	71.0	65.5	7.7	S.
	Buffalo	70.2	71.0	64.0	10.9	S. W.
	New York	73.5	71.0	67.0	9.1	S. & S. W
	Rochester	70.4	67.0	63.2	7.1	S. W.
N. Carolina	Charlotte	77.8	69.0	70.3	5.0	S. W.
	Raleigh	77.8	73.0	71.3	....	S. W.
	Wilmington	78.7	79.0	73.7	....	S. W.
	Asheville	72.2	....	....	4.7	S. E.
N. Dakota	Bismarck	70.2	61.0	61.2	9.0	N. W.
	Williston	69.2	56.0	59.2	9.0	N. W.
New Mex.	Santa Fé	68.7	46.0	55.2	6.1	N. E.
Ohio	Cincinnati	77.7	63.0	68.7	6.6	S. W.
	Cleveland	72.5	70.0	65.8	11.7	S. E.
	Columbus	75.0	64.0	66.5	8.7	S. W.
	Toledo	73.7	65.0	65.7	8.0	S. E.
Oklahoma	Okla. City	79.0	64.0	70.0	9.0	S.
Oregon	Portland	66.3	60.0	58.0	7.9	N. W.
	Roseburg	66.1	57.0	57.0	4.0	N. W.
Penna.	Erie	71.8	69.0	64.8	9.0	W.
	Philadelphia	75.8	66.0	67.8	9.4	S. W.
	Pittsburgh	74.6	68.0	67.1	5.2	S. W.
	Scranton	71.8	68.0	64.5	6.1	S. W.
	Harrisburg	74.5	67.0	67.0	5.7	W.
S. Carolina	Charleston	81.3	76.0	75.3	9.8	S. W.
	Columbia	81.1	72.2	74.1	7.0	S. W.
	Aiken	81.3	65.5	72.5	....	....
S. Dakota	Huron	71.6	65.0	63.6	10.4	S. E.
	Pierre	75.0	51.0	63.0	9.6	S. E.
	Rapid City	71.9	47.0	59.5	7.5	W.
	Yankton	74.6	66.0	66.8	6.5	S.
Tennessee	Chattanooga	77.8	71.0	70.8	5.2	S. W.
	Knoxville	76.2	74.0	70.2	5.0	S. W.
	Memphis	80.7	68.0	72.7	7.4	S. W.
	Nashville	79.4	68.0	71.4	5.0	S. W.
Texas	Abilene	82.2	52.0	69.2	9.0	S.
	Corpus Christi	81.9	80.0	76.9	12.0	S. E.
	El Paso	80.5	44.0	64.5	9.7	W. & E.
	Galveston	83.0	76.0	77.0	10.0	S.
	San Antonio	82.4	65.0	73.4	7.0	S. E.
	Ft. Worth	82.5	51.0	69.5	10.0	S.

TABLE 92.—EXTREME WEATHER CONDITIONS.
(JULY AVERAGES.)—(Concluded.)

State	City	Dry bulb in Deg. Fahr.	Relative Humid-ity in Per Cent.	Wet bulb in Deg. Fahr.	Wind Miles per Hour	Direction of Wind
Vermont	Northfield	66.1	78.0	61.6	....	S.
	Burlington	68.2	76.0	63.2	....	S.
Virginia	Lynchburg	77.3	69.0	69.8	....	S. W.
	Norfolk	78.4	74.0	72.6	....	S. W.
	Richmond	79.2	70.0	72.0	....	S.
Washington	Seattle	63.3	64.0	56.3	5.5	W.
	Spokane	68.8	41.0	55.0	....	S. W.
	Walla Walla	74.3	40.0	59.0	....	S.
W. Virginia	Parkersburg	74.9	68.0	67.5	4.0	W.
Wisconsin	Green Bay	69.6	68.0	62.6	8.0	N. & S.
	La Crosse	72.6	71.0	66.0	6.0	S.
	Milwaukee	69.7	77.0	64.7	9.8	N.
Wyoming	Cheyenne	67.4	46.0	54.6	8.0	N. W. & S.
	Lander	66.6	43.0	52.6	4.0	S. W.
Porto Rico	San Juan	79.9	80.0	75.2	10.0	E.
Cuba	Havana	80.5	81.5	76.6	10.0	E.
	Santiago	80.2	74.0	73.9	5.4	N.
Jamaica	Kingston	80.3	70.0	73.0	6.3	N. E.
Canada	Montreal	69.5	79.0	65.0	11.3	S. W.
	Toronto	68.5	70.5	62.3	7.9	N. W.
	Winnipeg	66.0	77.5	61.0	11.3	N. W.
	Calgary	60.6	65.6	53.6	7.5	N. W.
	Victoria	60.0	69.0	54.0	9.5	S. W.
Bermuda	Hamilton	78.7	84.0	74.7	9.5	S. E.
England	Durham	59.5	79.0	56.0	3.5	S. W.
	Sheffield	60.7	79.0	57.0	3.5	W.
	Birmingham	60.4	75.0	56.0	3.5	W.
	London	62.8	71.0	57.0	3.5	S. W.
Scotland	Glasgow	58.0	77.0	54.0	3.5	W.
Ireland	Dublin	60.5	77.0	56.0	3.5	S.
	Armagh	58.4	....	....	3.5	....
China	Hongkong	81.9	82.0	77.7	5.1	S. E.
	Macao	79.0	83.7	75.3	9.9	S. E.
	DATA BELOW TAKEN IN JUNE					
Brit. Ind.	Bareilly	89.8	58.0	78.0	3.1	S. E.
	Delhi	93.2	45.0	76.5	3.6	S. E.
	Bombay	82.4	84.0	78.4	11.1	W. S. W.
	Bhavnagar	89.3	71.0	81.3	11.5	S. W.
	Calcutta	84.8	80.0	79.8	4.3	S.
	DATA BELOW TAKEN IN DECEMBER					
Chile	Valparaiso	65.1	68.0	58.6	5.0	S. W.

cooling effect are the lowering of the temperature of the air and the condensation of some of the moisture contained in the air.

The spray type air washer which cools the air by passing it through a fine spray of cold water was widely used in the early installations of air conditioning. As efficient types of fin coils were developed and placed on the market at moderate prices they came into wide use on the smaller jobs. The range of application of the coils continued to expand into the larger jobs until the greatest portion of all comfort air conditioning is handled by coils.

For a number of years all coils were cooled with water or brine but with the development of Freon 12 it came into use by direct expansion into the coils on many jobs. In centrifugal compressors however the suction side with Freon 11 or similar low pressure refrigerants operates at a rather high vacuum and it is necessary to keep pressure drop between the expansion valve outlet and the machine suction at a minimum. Therefore water or brine is still used as a secondary refrigerant with refrigerants of this type.

Air Conditioning for Human Comfort.—The proper cooling and conditioning of air for human comfort in such places as theatres, restaurants, churches, office buildings, auditoriums, and homes has made rapid progress during the last few years. Various types of systems, some using mechanical refrigeration, others using ice, have been developed and perfected for this class of work.

In the following paragraphs special consideration will be given to standards of ventilation for comfort, sources of heat in air cooled and conditioned rooms, apparatus, and the application of the same.

General Requirements.—Proper air conditioning for human comfort depends upon a number of factors, the most important of which are the following:

1. Temperature of air.
2. Relative Humidity.
3. Air Motion and Speed.
4. Volume of air induced into conditioned space.
5. Purity of air.
6. Distribution of air.

The first three factors, namely temperature of air, relative humidity, and air motion have been used to develop an arbitrary standard of comfort measurements known as "effective temperature." (Fig. 173.) This is an experimentally determined scale which is a true measure for one's comfort for various combinations of air temperatures, relative humidity, and air motion.

Data on this subject presented here is the result of the work done

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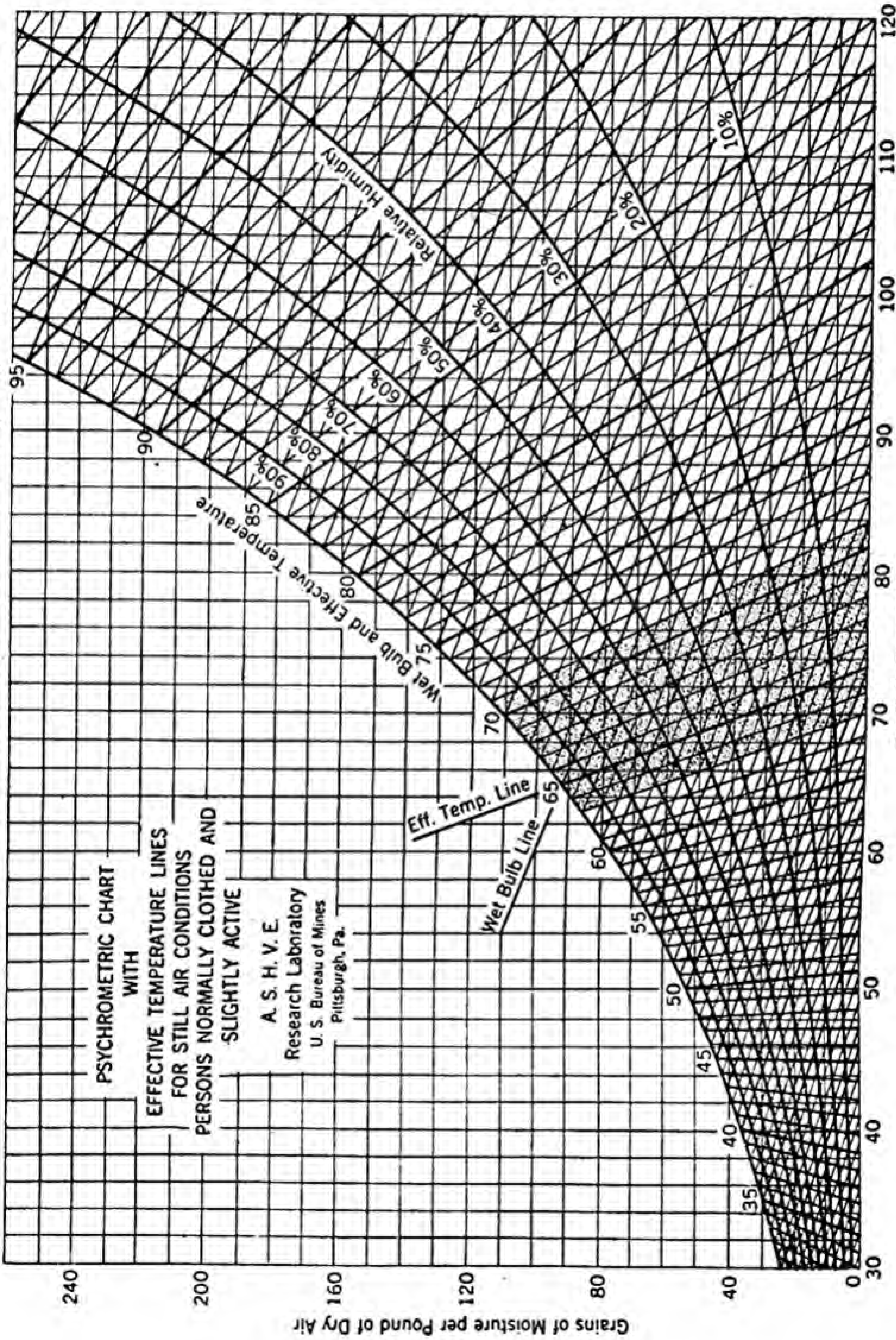

Fig. 173.—Psychrometric Chart with Effective Temperature Lines for Still Air. (Shaded Area Indicates the Comfort Zone.)

at the A. S. H. V. E. Research Laboratory in cooperation with the U. S. Bureau of Mines and it appears in the A. S. H. V. E. Guide, 1932.

These figures are really psychrometic charts of the Carrier type, upon which lines of equal or constant effective temperatures have been drawn. The shaded area gives the range of effective temperatures under which 50 per cent of the people feel comfortable, according to the conditions shown on the chart.

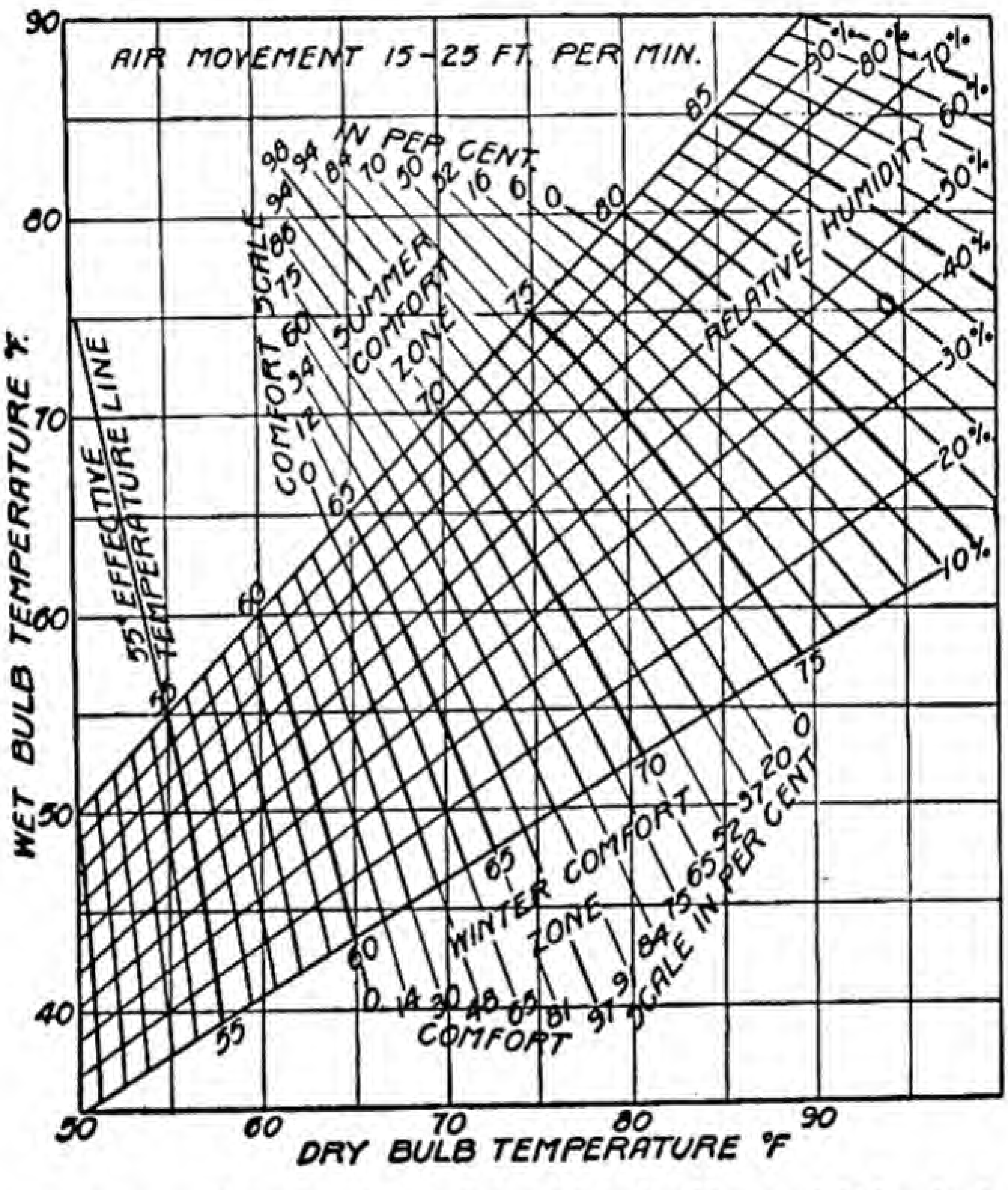

Fig. 174.—Comfort Chart Issued by the Harvard School of Public Health.

Where buildings are artificially cooled for human comfort during the summer, the inside temperature depends to a large extent upon the outside temperature. In other words, the inside temperatures should increase in some relation to the outside temperature.

Table 93 gives the desirable indoor temperatures in summer corresponding to various outdoor temperatures, and is based upon average moisture conditions outside, and upon relative humidities of 50 to 60 per cent on the inside.

The relationship of optimum conditions for both summer and winter are shown in Fig. 174. This chart shows that in the summer time the feeling of maximum comfort lies along the 71° F. effective temperature line. In winter time the feeling of maximum comfort lies along the 66° F. effective temperature line.

TABLE 93.—DESIRABLE INDOOR TEMPERATURES IN SUMMER CORRESPONDING TO OUTDOOR TEMPERATURES.

Degrees Outside	Degrees Inside		
Dry Bulb	Dry Bulb	Wet Bulb	Effective Temperature
95	80.0	65.2	73.4
90	78.0	64.5	72.2
85	76.5	64.0	71.1
80	75.0	63.5	70.2
75	73.5	63.0	69.3
70	72.0	62.5	68.2

Reprinted from A. S. H. V. E. Guide, 1932.

TABLE 94.—HEAT EMITTED BY PERSONS PER HOUR AT DIFFERENT ROOM TEMPERATURES.

H = Heat emitted by man at rest per hour.
Hl = Heat emitted by man at light labor per hour.
Ha = Heat emitted by man at average labor per hour.
Hh = Heat emitted by man at hard labor per hour.

$$H\ E = \text{Heat Energy} = \frac{\text{Foot-Pounds per hour}}{778} = 84 \text{ Btu., } 168$$

Btu. and 252 Btu. respectively for light, average and hard labor.

T = Room Temperature.

$$H = 13.2\ (98.6 - T) \quad \text{Heat due labor} = \frac{T \times H E}{100}$$

$$Hl,\ Ha,\ \text{or}\ Hh = 13.2\ (98.6 - T)\ \text{plus}\ \frac{T \times H E}{100}$$

Room Temp. Deg. F.	Rest	Heat Emitted by Man * Btu. per Hour at: 84 Btu. Light Labor	168 Btu. Average Labor	252 Btu. Hard Labor	Condition Required to Balance Excess and Shortage in Heat Emission
30	905	931	954	981	Increasing Humidity.
40	773	807	838	874	Heavy Clothing for Reduction
50	642	684	723	768	or Prevention of Radiation.
60	509	559	606	660	
68	404	461	518	575	Normal Condition.
70	378	436	491	554	
75	312	375	438	501	Decreasing Humidity.
80	246	313	375	447	Air Currents for Producing
85	180	251	322	394	Evaporation of Perspiration.
90	114	189	259	342	

* For children use one-half of table values.
Reprinted from A. S. H. V. E. Guide, 1932.

Sources of Heat in Air Conditioned Rooms.—The most important sources of heat in air conditioned rooms are as follows:

1. Heat emitted by occupants.
2. Heat transmitted through walls, roof or floor of room.
3. Heat generated by lights or motors.
4. Infiltration of outside air.
5. Sun effect.

Heat Emitted by Occupants.—The heat emitted by occupants constitutes one of the largest sources of heat generation within air conditioned rooms. Table 94 gives the heat emitted per person per hour at different room temperatures and at different rates of work.

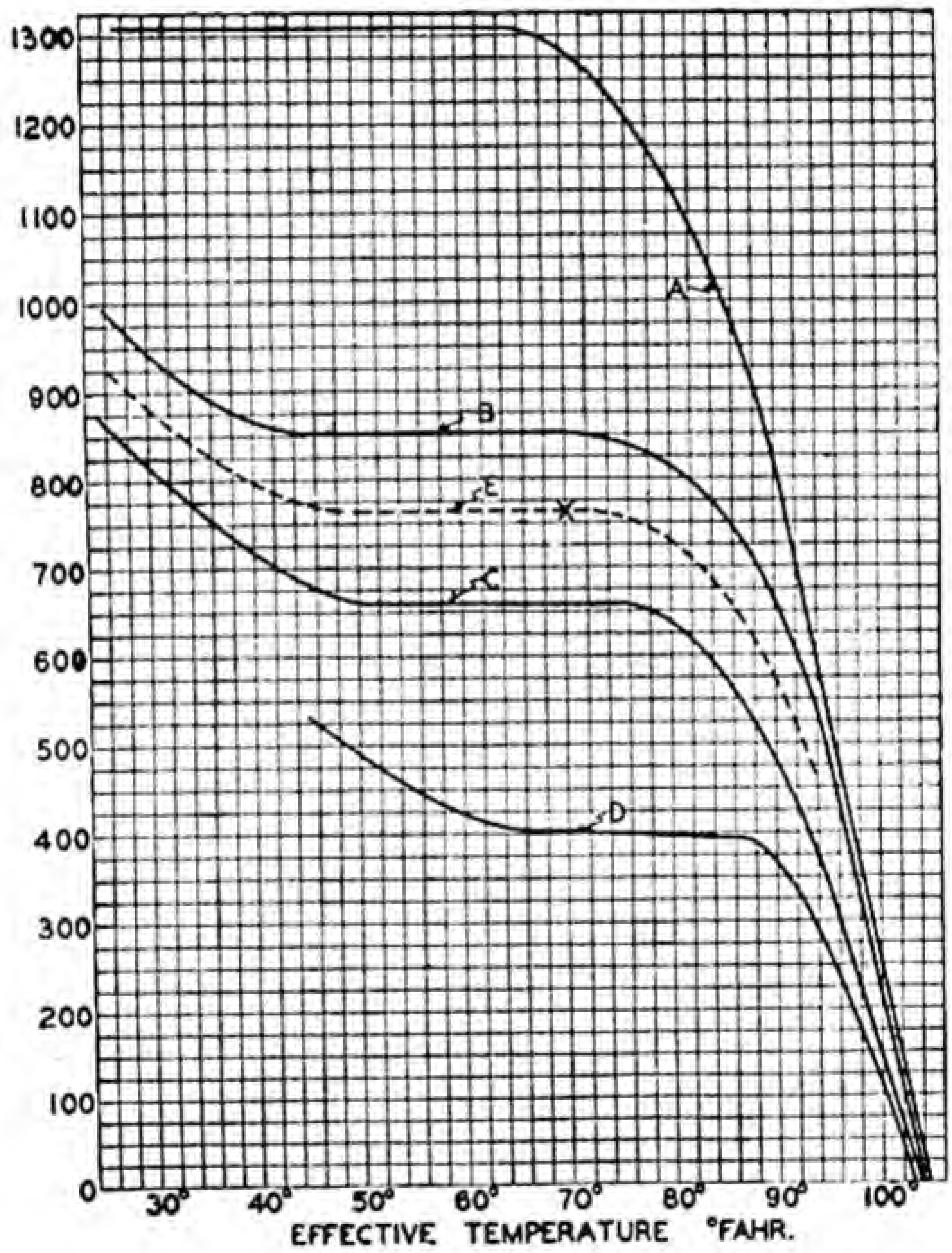

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Fig. 175.—Relation Between Total Heat Loss from the Human Body and Effective Temperature from Still Air.*

* Curve A—Men working, 66,160 ft.-lb. per hour. Curve B—Men working, 33,075 ft.-lb. per hour. Curve C—Men working, 16,538 ft.-lb. per hour. Curve D—Men seated at rest. Curves A and C drawn from data at an effective temperature of 70 deg. only, and extrapolating the relation between curves B and D, which were drawn from data at many temperatures.

The heat given off by the human body is really made up of two parts. The first part is the actual radiation and convection loss of heat from the body, and is sometimes termed sensible heat. The second part of the heat loss from the body is the loss by means of water vapor. This amount of heat is sometimes termed latent heat loss of the body.

The relationship between the sensible and latent losses of heat from the human body depends upon such factors as air temperaure, air velocity, etc. Figs. 175, 176, 177 and 178 prepared by the A. S. H. V. E. show the latest data on this subject.

Fig. 175 shows the relation between the total heat loss from the human body and effective temperature for still air. Fig. 176 shows

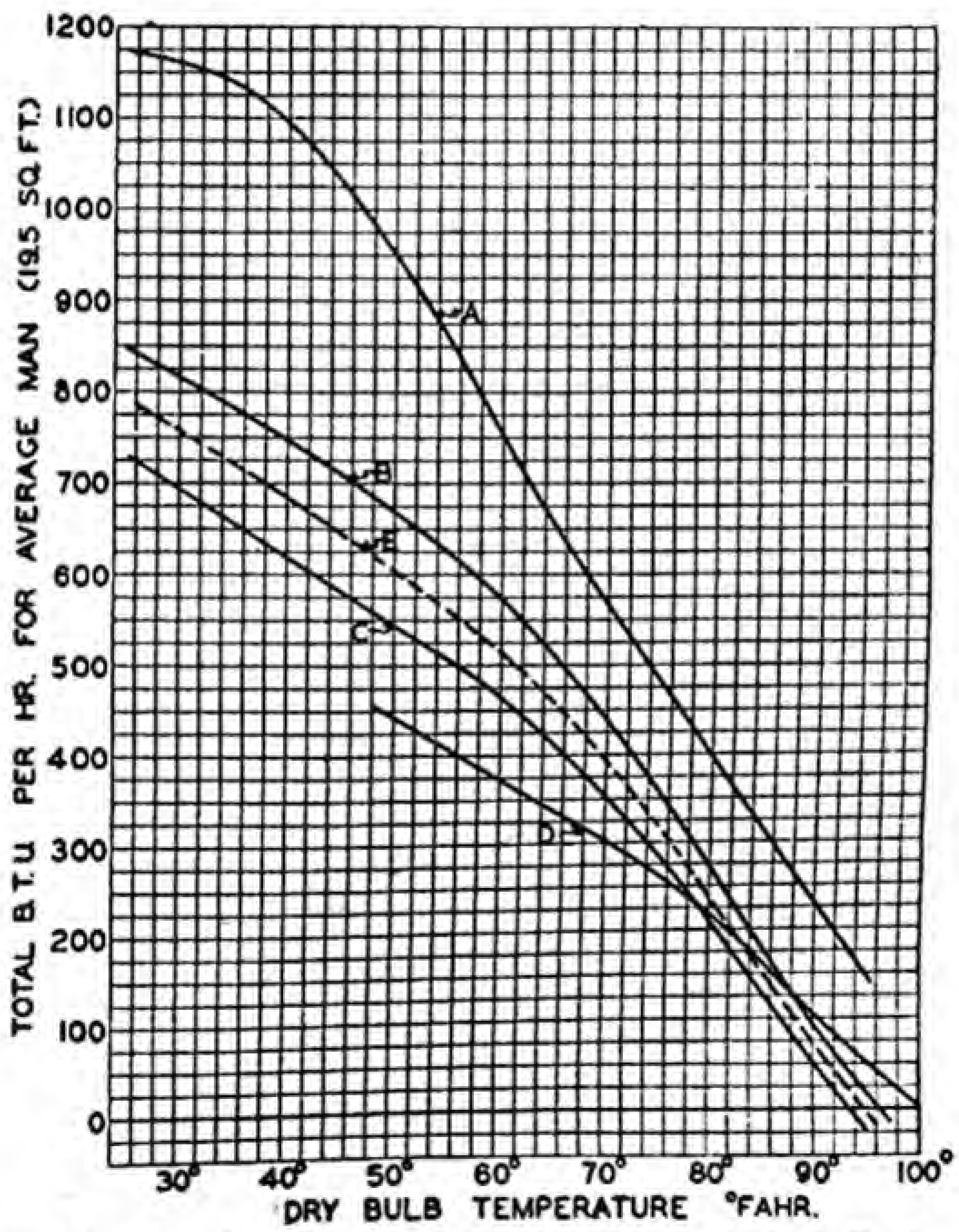

Fig. 176.—Relation Between Sensible Heat Loss from the Human Body and Dry-Bulb Temperature for Still Air.*

* Curve A—Men working, 66,150 ft.-lb. per hour. Curve B—Men working, 33,075 ft.-lb. per hour. Curve C—Men working, 16,538 ft.-lb. per hour. Curve D—Men seated at rest. Curves A and C drawn from data at a dry-bulb temperature of 81.3 F. only, and extrapolating the relation between curves B and D, which were drawn from data at many temperatures.

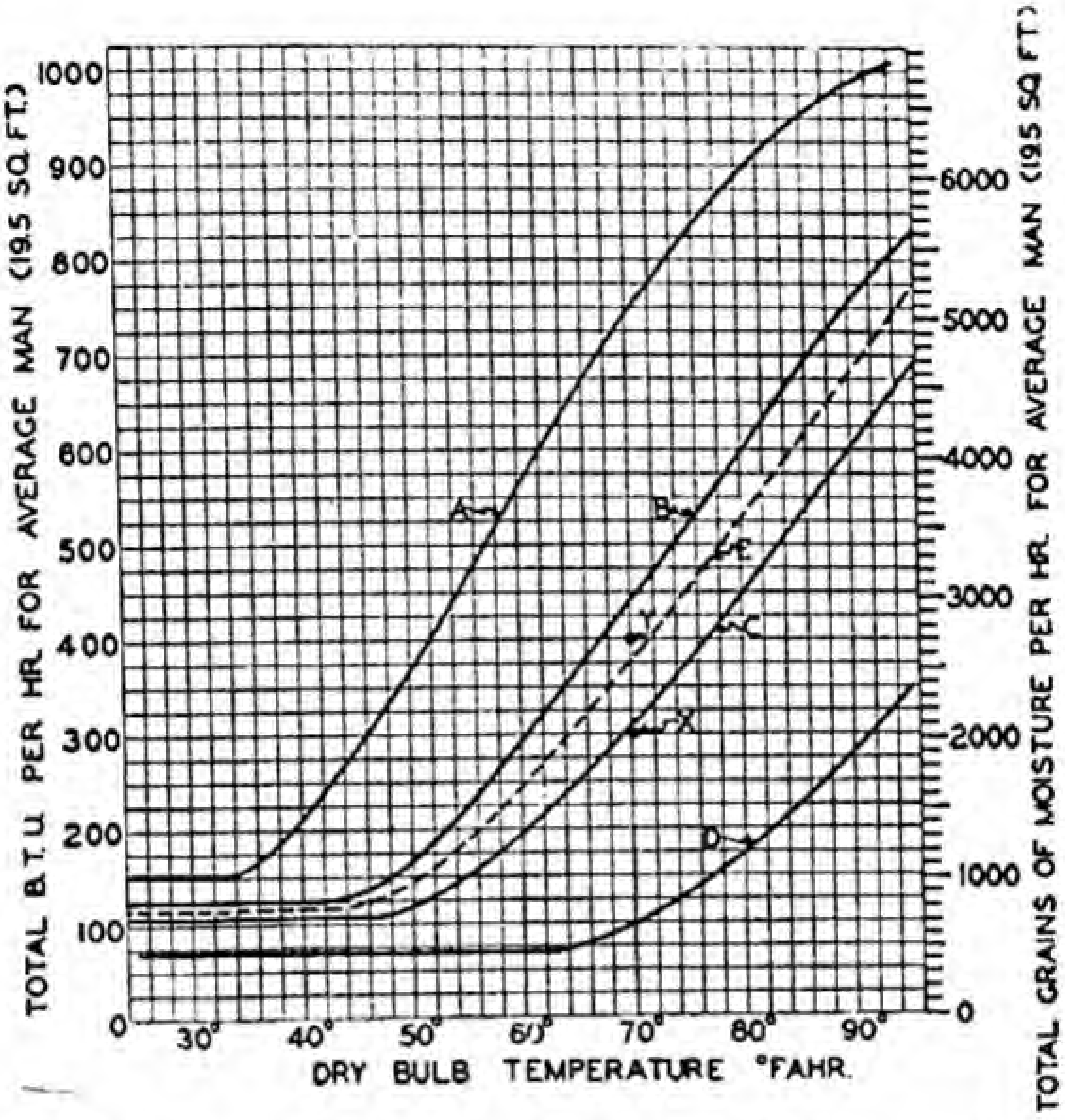

Fig. 177.—Latent Heat and Moisture Loss from the Human Body by Evaporation in Relation to Dry-Bulb Temperature for Still Air Conditions.*

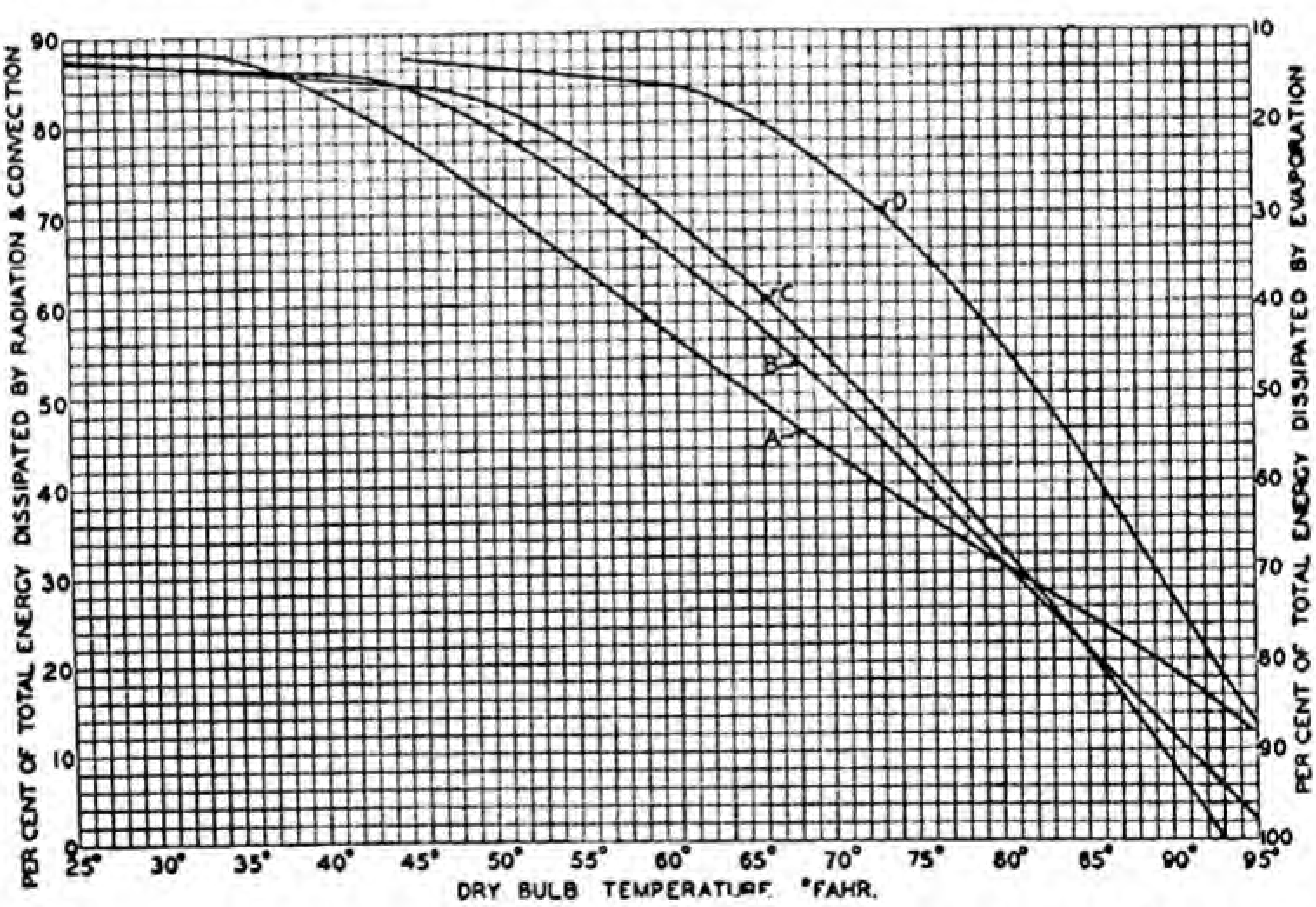

Fig. 178.—Heat Loss from the Human Body by Evaporation, Radiation and Convection in Relation to Dry-Bulb Temperature for Still Air Conditions.*

* Curve A—Men working, 66,150 ft.-lb. per hour. Curve B—Men working, 33,075 ft.-lb. per hour. Curve C—Men working, 16,538 ft.-lb. per hour. Curve D—Men seated at rest. Curves A and C drawn from data at a dry-bulb temperature of 81.3 F. only, and extrapolating the relation between curves B and D, which were drawn from data at many temperatures.

the relation between sensible heat loss from the human body and dry bulb temperature for still air, this sensible heat being made up of losses due to radiation and convection. Fig. 177 shows the latent heat and moisture loss from the human body by evaporation in relation to dry bulb temperature for still air conditions. Fig. 178 shows the heat loss from the human body by evaporation, radiation, convection in relation to dry-bulb temperature for still air conditions.

Heat Transmission Through Building Walls, Roof, or Floors.—The amount of heat transmitted into conditioned rooms depends upon kind, thickness, exposure, etc., of the walls, floors or roof. The general procedure for heat transmission calculations was given in Chapter VIII, and may be applied to any type of building construction.

Heat Generated by Lights and Motors.—The heat generated by electric lights will be at the rate of 3.415 Btu. per hour per watt. One Welsbach gas light will produce 2,100 to 3,000 Btu. per hour. One fish tail gas light will generate 3,500 to 5,000 Btu. per hour. Electric motors driving fans, pumps or other machinery will generate heat at the rate of 2,546 Btu. per horsepower or 3,415 Btu. per kilowatt, per hour.

Infiltration.—Losses due to infiltration will depend upon a number of local conditions. In the case of air cooled and conditioned rooms proper allowances must be made for this loss, which depends upon the amount of infiltration, outside conditions, inside conditions, number of openings and wind velocity. Under usual conditions air infiltration will correspond to one-half to three air changes per hour.

Sun Effect on Roofs and Skylights.[1] — The heat transmission through roofs and skylights will be increased due to the rays of the sun upon them. When the outside temperature is 80° to 95° F. then the outside roof temperature will be 135° to 150° F. at midday in July and August. Consequently, an allowance must be made for increased temperature difference. If roofs are sprayed with water, the effect of sun rays are not sufficient to be considered.

When the inside temperature is 80° F. protected skylights will transmit approximately 100 Btu. per square feet per hour. If white shades are provided on the skylights, approximately 60 Btu per square feet per hour will be transmitted. Vertical glass on west walls will transmit approximately 30 Btu. per hour per square feet.

[1] "Mechanical Equipment of Buildings" by Harding and Willard; Vol. 1, second edition; page 246.

Air Conditioning Applied to Human Comfort.—Where air conditioning, however, is applied to human comfort[1] we are interested primarily with the effect of the atmospheric conditions upon the people. They cannot be comfortable with drafts, especially if it happens to hit them in the back of the neck or around the legs. Since the purpose of "cooling for comfort" is to produce comfort, the method of air distribution to avoid sudden changes in temperature and noticeable drafts is the first problem to be met with that is not found in industrial applications.

The second problem not found in industrial applications is the greater quantity of air to be distributed per square foot of floor space in thickly occupied auditoriums. The greater the amount of air to be introduced, or rather the greater the amount of cooling to be performed in a given floor space, the greater is the difficulty of introducing that air without the introduction of drafts.

The third and perhaps the greatest difficulty which is present in cooling for comfort, but not found in industrial applications, is the practical necessity of concealing all duct work and making the outlets and method of distribution conform with the architectural scheme of the building and with the details of its decoration.

Theaters of all types, many with most expensively decorated ceilings, including the particularly "atmospheric" type of theaters in which the ceiling depicts the sky, all have been handled so as to give satisfactory distribution without in the least compromising the artistic requirements.

Two principal types of distribution have been found satisfactory for theaters and other auditoriums. One is the panel system in which the air is blown directly downward on a panel placed slightly below the general ceiling level. In large office buildings and banking rooms this deflector panel is sometimes incorporated with the lighting fixtures.

The other type used principally in theaters is known as the ejector system. In this the air is blown through a series of nozzles placed on the ceiling in the rear of the auditorium and directed toward the stage. The success of this system depends upon the fact that cold air blown horizontally into the room near the ceiling carries with it three to four times as much warm air by induction effect. This induction effect is what produces the return circulation. The cold air is thoroughly diffused and passes backwards in a slight uniformly distributed current over the entire lower half of the auditorium.

This system, however, has to be used with the greatest care because the distance of the blow, the size of the discharge nozzles, their

[1] W. H. Carrier, *Ice and Refrigeration*, March, 1931.

spacing, and the velocity used must all be definitely related to secure the definite results desired. This relationship has been worked out by experimental research, made, first in the laboratory and second in the field, so that now the design can be relied upon with confidence in almost any situation where such application is permissible.

Both of these systems are of the type commonly termed overhead systems; that is, air is introduced at or near the ceiling, where the cold air is mixed with the warm air of the room before it is brought into contact with the occupants. Such a type of distribution is logical and has been found necessary wherever cold air is used for ventilation purposes.

Upward circulation with distribution at the floor line has been found an utter failure. This method of distribution for ventilation in years past was the favorite one with the ventilating engineer and indeed it had become a standardized method of ventilation procedure when cold air was introduced at 10 to 15° F. below the room temperature.

The general public impression has been, and is even today, that air conditioning for comfort amounts to nothing more than cooling air by refrigeration. Air in an auditorium that is merely cooled has a high degree of saturation and is not only exceedingly uncomfortable, but probably unhealthful. People entering from a hot, moist outside atmosphere are covered with perspiration and cold damp air serves merely to condense this vapor in the clothing, and gives the feeling of a cold damp cellar. The purpose of an air conditioning installation should be to produce in the auditorium air that is reasonably dry, but with not too low a dry bulb temperature. The relative humidity, in fact, should be between 45 and 55 per cent with temperature as high as 78 or 80° F. in hottest weather, a condition under which a person will not feel a shock entering from the outside, yet the perspiration will be gradually removed from the body and from clothing.

Many early attempts at auditorium cooling were made not only with an upward system of ventilation, which was wrong, but without any regard to the control of the relative humidity. The air was introduced practically saturated, further moisture was added from the bodies of the people. With such applications it is no wonder that cooling for comfort, at least in theaters, became almost discredited. In industrial applications it either had been found necessary to reduce the volume of air supplied or to add heat artificially as through steam coils to control the relative humidity. The cost by this method when cooling for comfort is prohibitive because of the large amount of refrigeration required.

Today the system used in cooling for comfort in auditoriums, department stores and other places of human occupancy is one in

which the humidity can be controlled independently of the cooling effect so that definite conditions, both of temperature and of humidity, are produced in an auditorium whatever the external variations of heat or of human occupancy. This is accomplished without varying the volume of air circulated and without the addition of any artificial outside source of heat. The air introduced into the auditorium need never be below 65° F., thus avoiding cold drafts. This result is obtained through an invention of L. L. Lewis, in which he provides for two streams of air; one which enters the air conditioner from the outside, where it is cooled by refrigeration to a relatively low dew point; the other stream is taken directly from the room, preferably through dry filters. These two streams of air are then mixed and introduced into the room through suitable overhead diffuser outlets.

The regulation of the cooling effect to varying requirements is obtained by varying the proportion of dehumidified air to that of recirculated air by a combination of dampers. The relative humidity is controlled by fixing the temperature to which the air passing through the conditioner is cooled and saturated. This temperature will always be substantially lower than the temperature at which it is permissible to introduce air into the room. In general the air passing through the air conditioner is cooled from 20 to 25° F. below the room temperature, thus insuring the desired relative humidity, while the temperature at which the air was introduced into the room, therefore the cooling effect, was entirely dependent upon the proportion of air recirculated to the air cooled.

Air conditioning of the smaller space in office buildings presents a somewhat different problem. Practically an individual control of each office is required in cooling just as required in heating. In addition to this, rearrangement of offices may interfere seriously with any stereotyped system of distribution. In practically all offices the space underneath the window is always available for air distribution as well as for heating. A form of cabinet which can either be concealed beneath the window or set out slightly from it seems to be the most satisfactory solution to this problem. The cold air supply and recirculation can be controlled directly at this point, as can also the heating in winter. This gives each room an individual control.

The manufacturers of mechanically cooled household and commercial refrigerators have entered the cooling field and several types of room coolers, designed as general purpose units for all small or medium size rooms, such as bedrooms, living rooms and other rooms in the home, small offices, directors rooms, restaurants, candy factories, beauty and barber shops, dress shops, and a wide variety of other uses are on the market.

While these units are not large they are compact and it is claimed powerful enough to bring room temperatures down to a point where cool comfort is assured even on the hottest day.

In one of these room coolers warm air is drawn in at the back, passed through a large cooling coil and then forced out into the room. Through this process air is cooled, dried and circulated at the rate of 650 cu. ft. per min. without annoying drafts and breezes. An adjustable thermostat automatically regulates the room temperature. The refrigerant is controlled by a thermostatic expansion valve. These room coolers are furnished in two different models.

Another company manufacturing household and commercial refrigerating units has put on the market a blower-type room cooler and standard condensing unit. It consists of a large area cross-fin coil mounted in a metal housing with adjustable air deflectors at the front and a special type exhaust fan at the rear; a special expansion valve, removable drip pan at the bottom, with connections for carrying away condensation, and with four lugs at the rear for mounting. At the front of the cooler are fully adjustable air deflectors which can be set at any angle by rotating the deflector rings and also by moving the deflectors themselves.

The fan is equipped with an alternating motor, induction type with no brushes. It is a 6-blade fan, 12 in. in diameter, runs at 1600 r.p.m. and has a net air delivery of approximately 525 cu. ft. per min. This cooler, it is claimed, consumes but 60 watts per hour.

Ice Refrigeration in Air Conditioning.—During the last few years, ice has been extensively used as a cooling medium in air conditioning work. Comfort cooling application requires a cooling medium at a temperature of from 45° to 50° F.

When ice refrigeration is used it takes the place of the complete mechanical refrigeration system including compressor, condenser, evaporator, etc. The rest of the air conditioning system, including the air washer in dry-type cooler, blowers, automatic equipment, etc., are required for either type of refrigeration.

Fig. 179 shows the refrigeration section of an ice system. One pump takes the cold water from the chamber under the ice and supplies the nozzles in the air washer in auditorium. The other pump takes the warmed water from the pan of the air washer and returns it to the ice melting chamber. Part of this water is sprayed over the ice and the rest is by-passed to the chamber beneath the ice. The temperature of the water supplied to the air washer is kept constant by means of a thermostatically controlled valve supplying the water to the sprays over the ice. The entire system can be made automatic in operation.

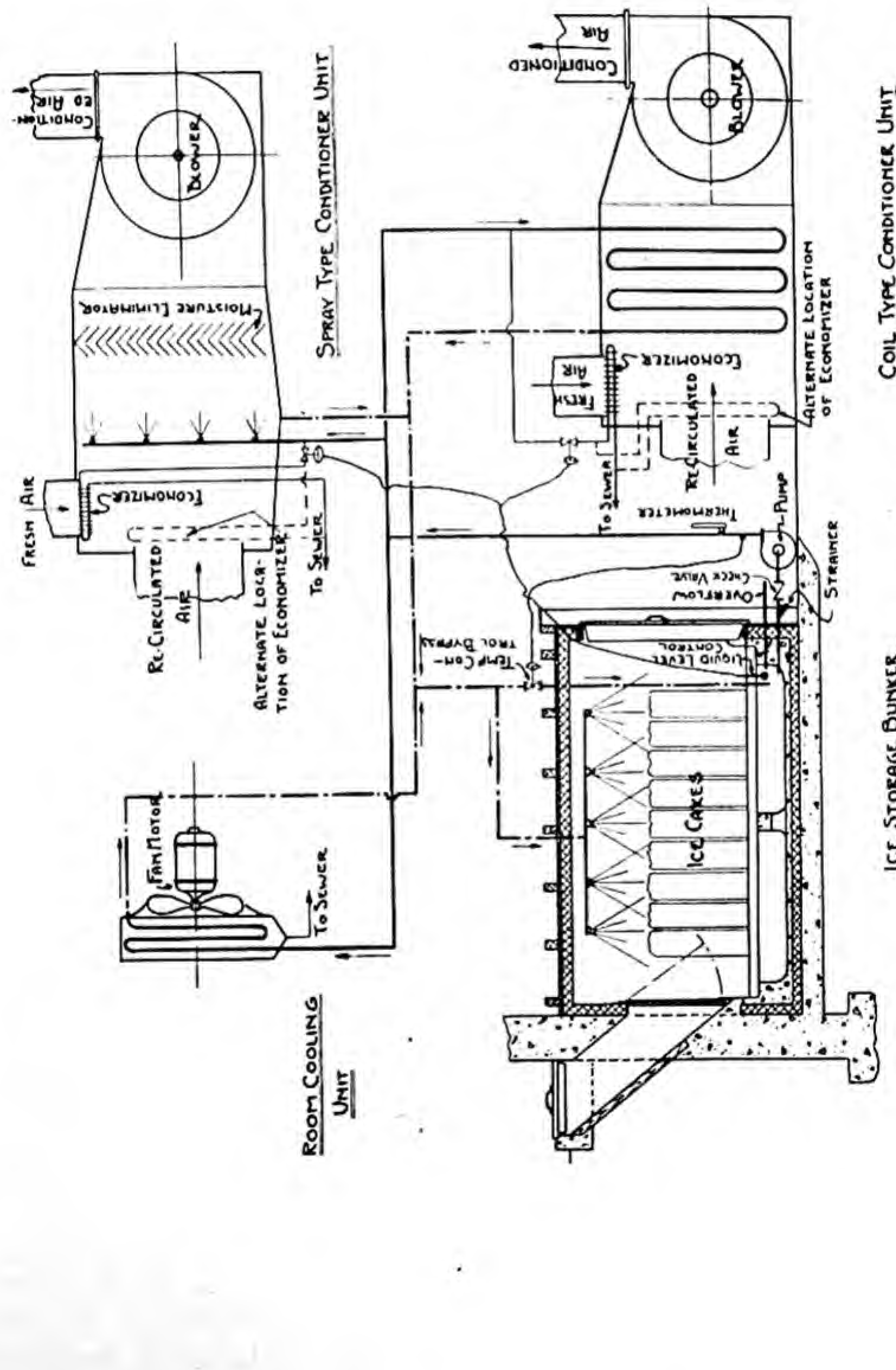

Fig. 179.—Typical Air Conditioning Installation Using Ice for Space Cooling.

The space required for ice refrigeration is usually much less than for mechanical refrigeration and the investment cost is materially reduced.

Ice refrigeration is particularly adaptable when the operating season is short, when the peak demand is high, and where the hours of daily use are of short duration.

Rate of Ice Meltage Determines Refrigeration Capacity. — One pound of ice will absorb about 165 Btu., due to its latent heat, and also to the resultant water being warmed to a point near the temperature of the air in the ice washer. It this takes place in a period of one minute it has the refrigeration rate of $\frac{165}{200} = .825$ tons of refrigeration in 24 hours. One ton of refrigeration requires melting of ice at rate of $\frac{12000}{165} = 72.7$ lb. ice per hour. If ice is melted at the rate of 2,000 lbs. per hour, it will have the refrigeration rate of $\frac{2000 \times 165}{12000} = 27.5$ tons refrigeration per 24 hours.

If the 2,000 lbs of ice is melted in 30 minutes the refrigeration rate will be $\frac{2000 \times 165}{6000} = 55$ tons refrigeration per 24 hours.

An air conditioning installation having a maximum requirement of 55 tons of refrigeration will require the meltage of ice at the rate of 4,000 lbs. per hour. It will be seen, therefore, that the capacity of ice refrigeration is very flexible. Almost any capacity may be obtained by accelerating the melting rate. Ice may be used in many types of equipment.

In order to obtain the maximum refrigeration from the ice supply, the water from the melting ice is not released from the system until it has been warmed to the highest possible temperature by cooling circulating air. Where a spray type air washer is used an economizer coil is frequently placed in the air stream ahead of the washer or in the entering stream of outside air. A valve controlled by the water level in the ice tank bleeds excess meltage off the circulating water line so that it runs through the economizer coil and out to the sewer.

In most ice air conditioning systems cooling of the water is accomplished by circulating the ice water through a coil. Usually the water flows through the coil counter-current to the air travel so that the water has reached a moderately high temperature at point of discharge

from the coil. In most cases of this type an economizer coil may not be justified and the excess water from the melting ice is released to the sewer by bleed off line near the main coil discharge.

Applications—One of the first extensive applications of ice air conditioning was in the field of railroad car cooling. Several thousand passenger cars and diners were equipped with ice air conditioning systems. Later the Association of American Railroads made an extensive investigation of all types of air conditioning systems for passenger cars and reported that for cooling seasons up to 5 months in length, the ice air conditioning systems had the lowest overall cost.

The ice systems have their best field in applications involving high peaks of refrigeration demand with a lesser number of total hours of operation. It is evident, therefore, that the system is especially well suited to use in churches and auditoriums. Neighborhood theatres which operate principally in the evening hours also find the ice system economical. This is also true of restaurants and night clubs where operations cover a limited number of hours.

Combined systems using cooling tower or well water do sensible cooling and ice water for dehumidification have proved quite effective in territories where the air is relatively dry or where shallow wells will provide water at around 60 to 70 degrees. The cooling tower or well water is circulated through the first coil in the air conditioner. Then ice water circulated through the second coil does the final cooling to whatever point is desired. Such systems have proved economical in operation. In fact many of them operate with 100 percent outside air and yet the cost of operation is not excessive.

QUESTIONS ON CHAPTER XVI.

1. In what industrial applications of refrigeration is air used as a refrigerating medium? Explain exactly how the air produces the desired refrigerating effect.

2. What is meant by the following terms: Dry air, saturated air, partially saturated air?

3. Define the following expressions: Humidity; absolute humidity; relative humidity; wet bulb and dry bulb temperatures; dew point.

4. Explain the relationship of wet and dry bulb temperatures, relative humidities, and dew points of air.

5. What is the volume of 100 lbs. of dry air at a pressure of 30 lbs. per square inch gauge and a temperature of 80° F.?

6. In a raw water ice plant, the air system delivers dry and saturated air at 30 lbs. per square inch gauge and at 40° F. Find the moisture content of the air in grains per cubic foot (one pound = 7,000 grains). What will be the moisture content of the air if it is expanded at constant temperature to a pressure of 15 lbs. per square inch gauge?

7. Air has a pressure of 14.0 lbs. per square inch absolute, and has a dry bulb temperature of 85° F., and a wet bulb temperature of 79° F. What is the corresponding relative humidity, dew point and moisture content? Determine the heat content of the air by two methods of calculations.

8. Atmospheric air, having a pressure of 14.5 lbs. per square inch absolute, a temperature of 90°, a wet bulb temperature of 83°, is to be cooled to dry and saturated air at 32°. Determine the refrigeration required per lb. of air, if all of the heat for cooling the air is removed by the refrigerating system.

9. Atmospheric air at a dry bulb temperature of 80° and a wet bulb temperature of 76°, is to be cooled and conditioned to a temperature of 60° and a relative humidity of 58°. The air is first cooled to remove the proper amount of moisture, after which it is heated to the desired temperature. Determine the initial cooling effect, in Btu. per lb. of air and the heat required to reheat air to the desired temperature.

10. Describe the different types of apparatus which are used for the cooling and conditioning of air.

CHAPTER XVII.

ERECTION AND TESTING OF APPARATUS.

Installation of Apparatus.—Having studied the fundamental principles underlying the operation of refrigerating machinery, having noted the principal types and construction of the various parts of the mechanical equipment, and having observed a few of the most important industrial applications of refrigeration, it is now opportune to give consideration to the installation and operation of the refrigerating apparatus. Thus, this Chapter will be devoted to the detailed consideration of the shipment and handling of equipment, the preparation of suitable foundations for the machines, the installation of the various piping systems, and the final inspection and testing of all parts of the equipment.

Shop Tests of Apparatus.—In the manufacture of mechanical equipment such as condensers, receivers, traps, valves, fittings, coils, coolers and compressors, the various component parts making up each piece of apparatus are carefully machined from the castings or formed from other material. The machine work is performed by accurate machines, the dimensions, locations, holes, etc., being determined by accurate jigs or templets. After the component parts are carefully formed in this manner the apparatus is assembled. The work of assembling the various parts of the mechanical equipment is generally performed by skilled workmen who have spent considerable time on each type of work.

Since the major portion of the mechanical equipment is designed to work under pressure in the refrigerating plant, the greater part of the equipment is tested under air pressure in the shops. Thus, receivers, traps, valves, fittings, coils, are usually tested under water at high pressure to make sure that they are mechanically perfect. In reference to the compressors it will be observed that it is a simple matter to test these out under air pressure by simply bolting blank flanges to all of the connections. The desirability of giving the compressor units a

running test in the shops seems to depend upon the size and type of the compressor, as well as the manufacturer. In some manufacturing shops, the compressors are given a running test. In other shops, only the static air pressure test is applied. The latter course may be due to the fact that the various component parts of the compressor are carefully machined and that the units are assembled by skilled workmen who have had considerable experience along these lines.

Shipment of Material.—As soon as the apparatus has been suitably tested in the shops, it is properly boxed and crated, after which it is placed in a railroad car for shipment. Generally each shipment is accompanied with a detailed material specification, giving the name of each piece, the quantity, and the box or crate number. This material specification is usually mailed immediately after shipment. After the shipment of the material has been received, it is advisable to check the material over closely with respect to the material specifications and the erection drawings so as to make sure that there is enough material to complete the plant. In addition, due to the fact that the shipper's responsibility ceases when the material and apparatus are delivered to the carrier, it is necessary to check the shipment before it is unloaded, against the bill of lading. By this means when material is lost or broken, proper claims for adjustment may be made. In the event that the part cannot be found or that it cannot be satisfactorily repaired, the erecting engineer should order duplicate parts.

Location of Plant and Apparatus.—In the erection of a new ice making or refrigerating plant, it is evident that detailed considerations should be given not only to the general location of the plant but also to the location and arrangement of the apparatus in the plant. The type of the installation will in general indicate approximately in what location the plant should be placed. The plant should be placed in the most desirable location available. Ordinarily this proposition is not given enough attention, or it is determined by persons who have not sufficient knowledge or practical experience to select a desirable location. The most suitable location for the plant should be the one that would serve best those who make use of the plant. In addition, consideration should be given to the facilities for handling the material required in the plant, such as fuel and water. The proper location of the plant and the efficient arrangement of the parts of the plant are so important that the ultimate financial success of the enterprise may be determined by these considerations. It is probably beyond the province of the erecting engineer to assist in the selection of a suitable location for the plant. On the other hand, it is often within his power to provide an efficient arrangement of the various parts of the mechanical equipment in the plant. In most all cases, the erecting en-

gineer will be supplied with detailed erection drawings showing the location and position of all parts of the equipment, but in many cases, due to the fact that he is "on the job" and therefore able to visualize in a more practical manner the layout of the plant he is sometimes in a position to rearrange the apparatus for better results.

The requirements that should be considered in the location of the apparatus are the sanitary conditions in the rooms, lighting, ventilation, space requirements for inspection and adjustments, the relation of auxiliary units to the main units, the safety of the operators, etc. Units of apparatus and their auxiliary parts should be located as close together as conveniently possible. The various units of the plant should be located close together so as to reduce as much as possible the length of the connections between the various units.

Foundations for Machinery.—One of the most important things to be considered in the erection of a refrigerating plant is the proper construction of the foundations for the machinery. It is evident that the foundations must have ample size, the right proportions, and be made from first-class materials in a workmanlike manner. The principal function of the foundation is to support the weight of the machine by distributing the pressure due to the weight over sufficient area so that the foundation will not settle and that the machine may be rigidly located. To fulfill the foregoing requirement, the foundation must have considerable weight and extend a considerable distance below the surface of the surrounding ground. This is necessary to eliminate the effect of the possible freezing and thawing of the ground and vibrations or loads which may be imposed upon the ground near the foundation of the machine. The depth of the frost line, of course, will be determined by the general location of the plant. In the colder regions the frost will extend six feet below the surface. In many cases, the local conditions in the plant will affect the design of the foundation.

The second function of the foundation is to absorb the forces which are produced by the rotating and reciprocating parts of the machine itself. The forces produced by the reciprocating parts of the machine act along the center line of the piston rod, while those produced by the rotating parts act radially in all directions from the center of the crank of the piston compressors and the like. In general, these forces may be said to depend upon the speed of the machine and the weight of the parts. These forces will generally cause a perceptible vibration of the machine and the foundation, if the foundation does not have sufficient weight. In most cases, the frame or bed-plate of the machine has considerable weight, which helps to absorb the forces due to the moving parts, but it is generally advisable to provide a foundation which has a weight considerably more than the total weight of the unit.

Another function of the foundation is to hold the machine still against other unbalanced forces. These unbalanced forces may be produced by the pull of the belt in a belt-driven machine, or from other sources. In addition to the foregoing requirements, the allowable weight per square foot on the soil may have an important bearing on the design of the foundation. In case that the foundation rests upon solid rock, it is evident that the minimum amount of weight is necessary. On the other hand, if the foundation is to rest upon marshy or soft ground, it is evident that the forces must be spread out over sufficient area to prevent the settling of the foundation. Various city ordinances prescribe the unit load to be allowed upon the soil in their localities. The following table, taken principally from Baker's "Treatise on Masonry Construction," shows the supporting power of various soils in tons per square foot:

Rock—granite, etc., in hard compact strata	200 to ..
Rock—limestone, equal to best masonry	25 to 30
Rock—sandstone, equal to best brick masonry	15 to 20
Rock—broken and well compacted	7 to 20
Rock—soft and pliable as shale, equal to poor brick masonry	15 to 20
Hard pan—gravel and sand, well cemented with clay	8 to 10
Clay—thick beds and dry	4 to 6
Clay—thick beds and moderately dry	2 to 4
Clay—soft, wet, confined	1 to 2
Gravel—coarse and dry, well compacted and confined	8 to 10
Sand—dry, compact, well cemented with clay	4 to 6
Sand—clear and dry, confined in natural beds	2 to 4
Quicksand—marshy and alluvial soils, etc., confined	0.5 to 1

When the bearing value of the soil is quite low, it is sometimes advantageous to use piling. This is especially true if the unit is quite large. At present, wood, steel, and concrete piles are being used, piles constructed of yellow or red pine, oak, birch or beech being used more extensively than the others. In the event that piling must be used the safe bearing load in pounds for each pile may be determined from the following formula *(Engineering News)*:

$$L = \frac{2\,wh}{s + 1}$$

in which L = safe load in lbs.
w = weight of hammer in lbs.
h = fall of hammer in ft.
s = last penetration in in.

Material for Foundations.—Materials which are commonly used at the present for foundations of machinery consist of concrete and brick, principally. Concrete foundations are coming into extensive use in most all parts of the United States, due to the fact that they are cheap and durable. Bricks are used in some parts of the country for

building foundations where materials for concrete are expensive. When foundations are built from brick, suitable footings constructed of concrete or stone should be laid down before the bricks are placed. The materials used in concrete foundations should, of course, be the best of their kind. The stone should be broken into lumps that would pass through a two-inch ring. It should, of course, be clean and dry. The sand should be what is known as sharp; that is, it should be coarse and gritty. In reference to the cement, first-class Portland cement is recommended.

Various proportions are used for mixing concrete. A good concrete can be made by mixing one part of Portland cement, three parts of sand and five parts of broken stone or coarse gravel. This mixture is known as 1:3:5 mixture. Another proportion that is sometimes used is 1:2:4. Another mixture that is sometimes used is as follows: One part of fresh Portland cement, two parts sharp sand, one part coarse gravel, and four parts broken stone. Concrete foundations constructed of the foregoing materials will weigh approximately 150 lbs. per cu. ft. At present, the concrete foundations are molded into solid masses. It is evident that they must be thoroughly dry before any appreciable weight is placed upon them. The time required for drying will depend upon the size and layout of the foundation, but generally five to ten days will be necessary to allow the foundation to properly dry and set.

Foundation Templets and Drawings.—Templets are wooden frames which are designed to support the foundation bolts in their proper location during the construction of the foundation. The templets are suspended just above the foundation. Holes are provided for inserting the foundation bolts. The foundation bolts should be placed so that they will be at a suitable height after the foundation is completed. In addition to this, suitable forms should be placed about the foundation bolts so that a space varying from two to three inches will surround the bolt from the top to within a few inches of the bottom. When these forms are removed the space is available for any slight horizontal adjustment which may be required.

Fig. 181 shows the method of determining how high the bolts should extend above the foundation and how a suitable form may be built around the bolt. The templets should be rigidly supported and in line with the walls of the building. The templet generally has the center lines marked upon it; that is, the center line of the crankshaft and the center line of the cylinder, when the foundation to be installed is for a compressor. The erecting engineer should, in general, depend upon the contractor, architect, or supervising engineer to furnish him with a line for location of the machine. The erecting engineer should not be forced to go beyond the limits of the machine room in order to

determine the proper location of the machine. After the erecting engineer has been furnished with a line for determining the location of the machine, he may use any of several methods to properly locate the templet. One of the most simple methods for a horizontal compressor is as follows:

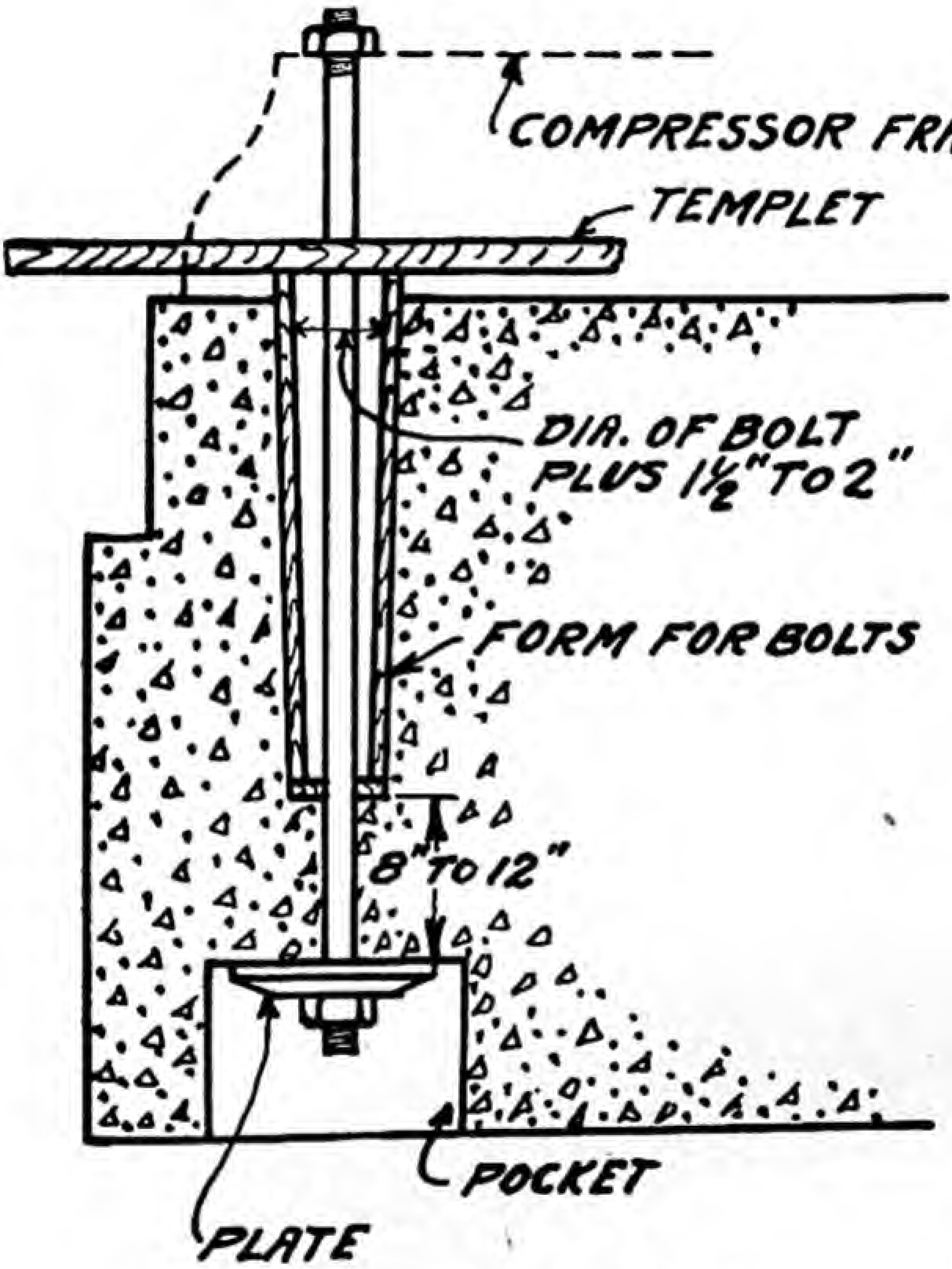

Fig. 181.—Foundation Bolt.

If the erecting engineer is supplied with a line which is parallel to the center line of the cylinder, it is an easy matter to lay out the center line of the machine parallel to the original line at the proper distance. After this, another line is laid out to represent the center line of the crankshaft. The line representing the center line of the crankshaft and the center line of the machine may be attached to the walls of the building, or any other stationary object. They should be as nearly horizontal as possible. In order to get the center lines of the shaft and

cylinder at right angles, the following method may be pursued, as illustrated by Fig. 182: From the intersection of the two center lines at O two points at equal distance on the center line of the crankshaft are laid off at points A and B. If the distance OA and OB are made equal to six feet, and the distance OC on the center line of the cylinder is

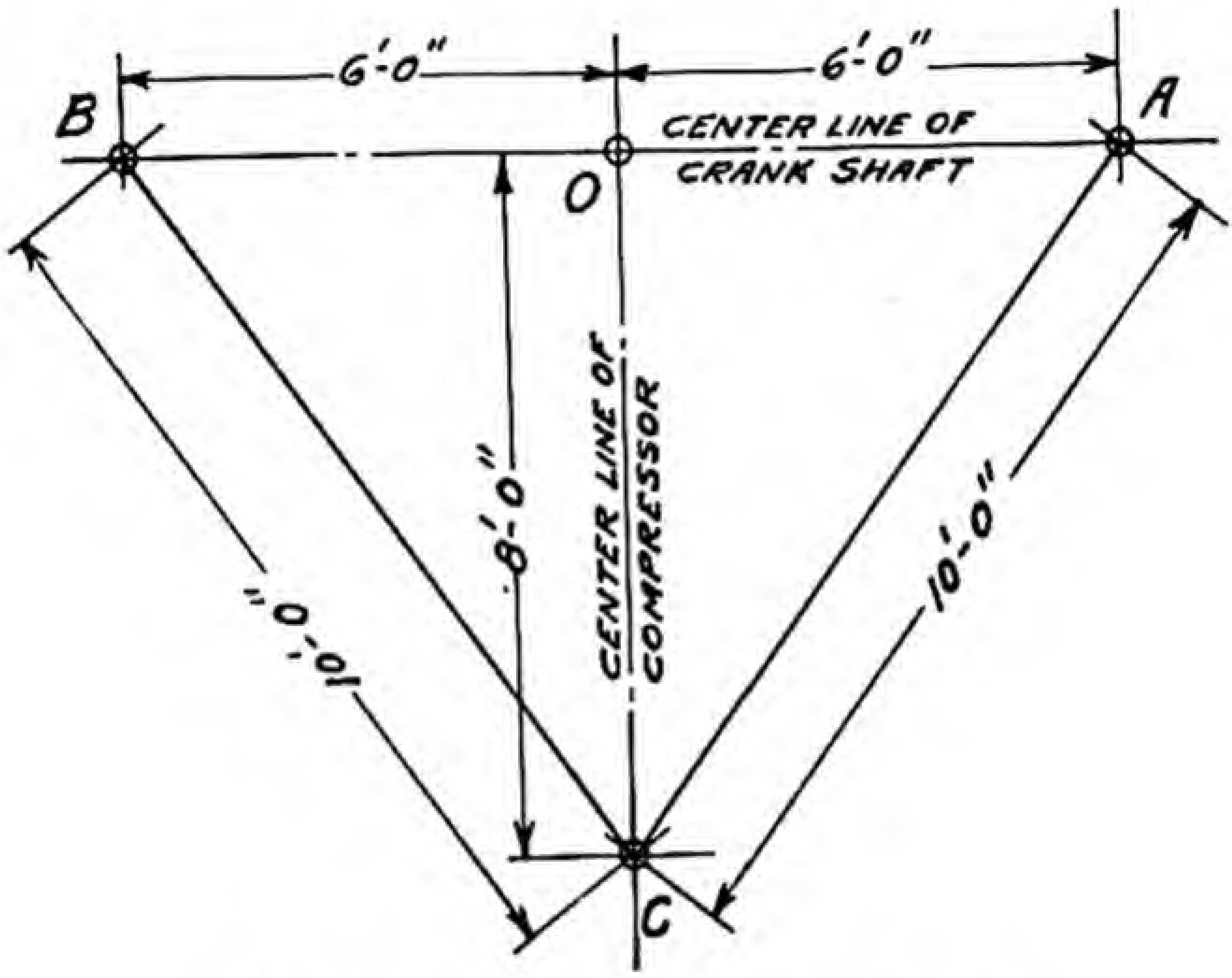

Fig. 182.—Location of Center Lines.

made equal to eight feet, than the length of AC and BC should both be exactly equal and in this case equivalent to ten feet. Probably more accurate measurements may be made by using 12, 16 and 20 feet for these distances, instead of 6, 8 and 10.

After the exact center lines of the crankshaft and cylinder have been determined in this manner, they may be transferred in locating the templet by means of plumb bobs. As previously indicated, the templet after being located in the foregoing manner should be firmly fastened in place.

The machine builder generally supplies a detailed drawing for the construction of the foundation. This drawing should show all the principal dimensions of the foundation, distances between the center line of the flywheels and the center line of the cylinder and the dimensions for locating exactly all of the foundation bolts. In addition, it should show the proper elevation of all points of the foundation, giving particular attention to the proper depth of the foundation and the amount of space to be allowed for grouting in the machine.

The manufacturer should supply, in addition to the foundation drawing, either a templet drawing showing the exact location of the foundation bolts in reference to the center line of the crankshaft and the center line of the cylinder, or provide a suitable wood templet which would automatically locate the bolts.

Foundation Bolts.—The purpose of the foundation bolts is to securely anchor the machine to the foundation. Thus the foundation bolts for small machines are seldom below ¾-in. in diameter, and on the larger compressors may have to be made as large as 1½ to 2 in. in diameter. They should be made of medium steel and have a considerable length of threads on both ends. The anchor plate on the lower end should be placed in the center of the threads so that some adjustment is available. They should not be made extraordinarily long on account of the effect of temperature changes; on the other hand, they should be made long enough to extend them to the concrete a considerable distance so that a considerable weight of concrete will be between the anchor plates and the frame of the machine.

The anchor plates on the larger sized bolts are generally made of cast iron and are designed in the manner of column bases. On the smaller foundations, the anchor plates may consist of simply a plate of metal, usually mild steel. In some cases, it may be necessary to place foundation bolts in solid rock, or in an old foundation. In this case it is necessary to drill a hole for the insertion of the foundation bolt. The shape of the hole to be drilled for this type of construction, together with the design of the foundation bolt, is shown by Fig. 183. The bolt has the lower end split so that a metallic wedge may be inserted. The method of inserting consists of starting the wedge into the split part of the rod before it is put into the hole. After it has been inserted, the rod is driven against the bottom of the hole so that the wedge enters into the split part of the rod, thereby expanding it. The construction is finished by filling the holes with cement and sand mortar.

Handling of Concrete.—After the templet has been properly located and the form constructed for retaining the concrete, the concrete material should be properly mixed and put into the form. In the event that the foundation is to be constructed of stone, sand and cement, the following method of procedure may be used: The proper proportions of sand and cement are put into one pile and thoroughly moistened with water. The crushed stone is put into another pile and thoroughly moistened with water. The sand and cement mortar formed in this manner is next shoveled on top of the pile of wet stone. These materials are next thoroughly mixed by the use of the shovel. In the event that the concrete is to be made from stone, coarse gravel, sand

and cement, the sand, gravel and cement should be put into one pile and then thoroughly moistened, after which the broken stone may be added and the whole thoroughly mixed. After the concrete has been mixed in this manner, it is ready to be placed in the molds. Any suitable means of transporting the concrete and depositing it into the molds may be used. The question that should be observed in filling the molds is to thoroughly ram the cement into place so as to exclude as far as possible all of the air.

Leveling and Grouting.—As previously indicated, the foundation should be so constructed that there is a space of one-half to one inch between the top of the concrete foundation and the bottom of the machine base. This allows space for leveling of the top of the foundation, and at the same time allowing the machine to be properly aligned. After the machine has been placed upon the foundation, it may be leveled and elevated to the proper height by driving wedges between the frame and the foundation. These wedges may be constructed of hard wood or iron. A suitable size is about six inches long, two inches wide and one inch thick, tapered to a point. The machine may be properly leveled by placing a spirit level on the crankshaft in one direction and on some other plane surface in the other direction. After the machine has been leveled and brought to the proper height in this manner, the space between the base and the foundation should be filled with a suitable grouting material. This usually consists of making a mixture of one part of clean sand and one part of Portland cement, after which enough water is added so that it will pour easily. The grout should fill all of the space around the foundation bolts between the frame and the foundation, and any other cavity directly beneath the base of the machine. The grouting should be allowed

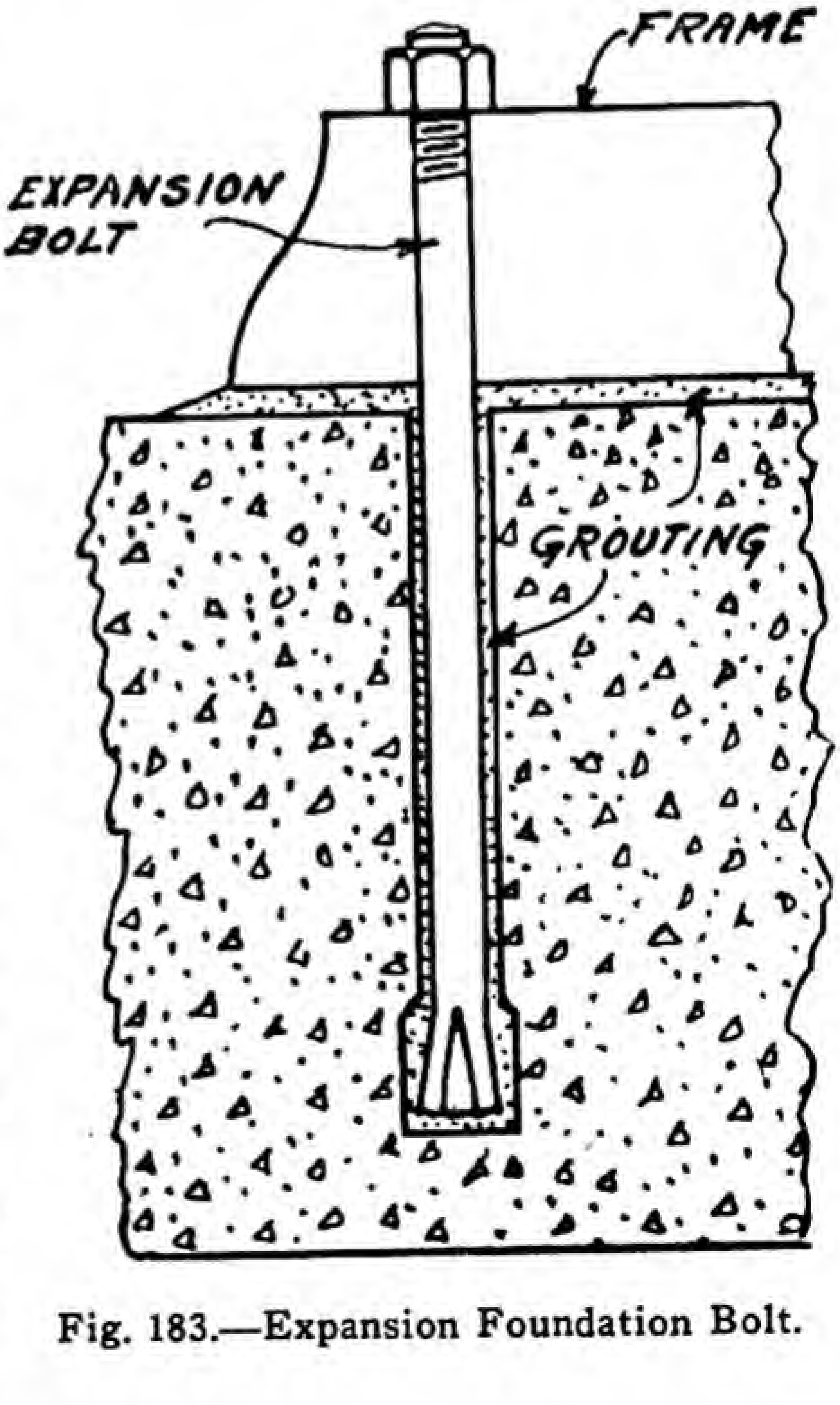

Fig. 183.—Expansion Foundation Bolt.

to dry for several hours, after which the outside and top part of the foundation can be smoothed up. When the grouting has been placed 36 to 72 hours, the foundation bolts may be tightened up. Cement grout for leveling and grouting in machines is used quite extensively at present. In some cases, other materials have been used, such as iron filings and sal ammoniac, sulphur, type metals, etc.

Pipe Connections.—After the various units of apparatus have been securely fastened upon their foundations, the various piping systems may then be completed. The typical refrigerating plant has a multiplicity of connections. Thus, there may be ammonia, steam and exhaust, air, brine, and water connections about the plant. It is obvious that each type of system must have its share of attention. Probably each erecting engineer will have his own individual method of erecting the various piping systems. This is not only true of the erection of the piping sysetms, but it is also true of the moving of the heavier pieces of the apparatus, such as flywheels, bed plates, shafts, etc. In addition, it is obvious that each particular plant will present new problems to the erecting engineer. Usually the erecting engineer relies on his own ingenuity in devising ways and means of moving the heavy pieces of machinery, the construction of rigging for heavy lifts, etc. However, it may be of interest to note some of the methods that are employed to make tight joints in the ammonia piping systems.

There are four principal means of making tight joints for ammonia connections which are used at present. These are the sweated soldered joint, the ordinary soldered joint, the litharge joint and the welded joint. The sweated soldered joint may be constructed in the following manner: The threads and the adjacent surfaces on the pipe and fitting should be thoroughly cleaned until they are bright. After this, the pipe and fitting are dipped into a pot of molten solder. This is usually composed of one part tin by weight and one part lead by weight. The primary purpose of dipping the fitting into the molten solder is to heat it. Otherwise, the pipe and fitting may be heated by other means, such as blow torches, etc. After the pipe and fitting have been heated, some soldering acid and sal ammoniac are put on the threads. The soldering acid may be made from muriatic acid which has been neutralized by the insertion of strips of sheet zinc in sufficient quantities to stop the effervescence in the acid. After covering the threads with the soldering acid the pipe and fitting are again dipped into the solder. The purpose of this is to thoroughly tin the threads and the surfaces adjacent to the threads.

The next step is to remove the fitting and pipe from the solder and screw the fitting onto the pipe tightly before the fitting cools. This makes a very tight joint and is really in the nature of a shrink fit. Par-

ticular attention should be given to the lining up of the bolt holes. The pipe and fitting should then be placed in a vertical position and the soldering recess filled.

The following instructions may be used for making a litharge joint: The threads on the pipe and fitting should be carefully cleaned, using gasoline or kerosene to remove the grease, dirt, or any other foreign matter which may be present. The litharge pipe cement may be made up by using one and a quarter parts litharge and one part glycerin by measure. Another formula for litharge cement is twelve pounds of litharge to one gallon of glycerin. The litharge and glycerin should be mixed thoroughly, and only enough made for immediate use. This is especially true in warm weather, since the litharge cement hardens or sets quickly. The joint is made simply by spreading a thin coating of the cement on the threads of both the pipe and the fitting, after which the fitting may be screwed onto the pipe tightly.

A simple soldered joint may be made as follows: The threads on the pipe are oiled, after which the fitting or flange is screwed onto the pipe, giving particular attention to the lining up of the bolt holes. The fitting and pipe are next heated in order to burn the oil off, and so that the fitting will supply heat for melting the solder. While the pipe and fitting are still at a high temperature, some soldering acid and sal ammoniac are put into the soldering recess. After this, the threads are thoroughly tinned by the used of a stick of solder. The stick of solder is applied continuously until the recess of the fitting is entirely filled, after which it may be filled up to a smooth fillet.

In the making of pipe-joints by any of the foregoing means, it is necessary to make sure that the threads are properly cut so that they will become gradually tight and will not shoulder at the end of the thread when the fitting is screwed onto the pipe. Brigg's standard pipe thread is used almost exclusively at present in the United States. All of the above methods of making pipe joints are in use at present. The soldered joint is probably used more extensively than the litharge joint. It is probably true that when either type of joint is perfectly made, the results will be satisfactory. The soldered joint is probably more permanent than the litharge joint, while on the other hand it may be observed that the litharge joint is more easily changed for making repairs or alterations.

Welding is being used more and more for making pipe connections. A good welder, when supplied with proper material, can connect the pipes quickly, accurately, and strongly, thus eliminating fittings, points where leakage may occur, etc. Welded pipe joints are not expensive when compared to the cost of fittings and the cutting of pipes and threads.

Welding is now being used extensively in the fabrication of headers,

receivers, accumulators, tanks and other pressure vessels, together with tanks. Both gas and electric welding are used; the particular one to be used depends upon the work to be welded and local conditions.

In respect to tools required for the purpose of erection, it is evident that each erecting engineer will have his own individual kit of tools, which will generally include tools for handling pipe up to two inches in diameter. The following tools will also be found useful in erecting the medium-sized type of refrigerating plant:

1 portable forge
1 large soldering pot
4 to 6 pieces of 2½-in. extra heavy pipe about 4 ft. long, not threaded
1 pipe threader to take 2-in. to 4½-in. pipe
1 pipe cutter to take 2-in. to 4½-in. pipe
1 pipe vise to take as large as 4½-in. pipe
1 set of stock and dies
2 large pipe wrenches to take up to 4½-in. pipe
2 large pipe wrenches to take up to 3-in. pipe

Inspection and Testing of Apparatus.—After all parts of the mechanical equipment have been thoroughly fastened on their foundations, and after all of the various pipe connections have been completed, it is desirable to go over each and every part of the equipment to check up the erecting work. All parts of the apparatus should be carefully examined before they are put into operation. All rotating and reciprocating units should be turned over by hand in order to make sure that nothing interferes with the moving parts. The erecting engineer should make sure that all the oil pumps, cups and lubricators are supplied with oil and that the oiling apparatus is adjusted for operation. Particular attention should be given to the kind of oil used. Oil which may get into the inside of the compressor through the stuffing box should be a good grade of ice machine oil. The packing and the stuffing box and other movable joints should be adjusted for operation. The packing of the piston-rod on the ammonia compressor should be given special attention, as this may become one of the sources of the largest ammonia leak about the plant. A packing especially designed for the operating conditions of the compressor should be used for this work. Metallic packing is being used quite extensively for this purpose at present. The machine room, especially, should be put into order. Tools, wrenches, oil cans and the like should be put in their respective places. This will lessen the danger of these falling among the moving parts of the machinery.

The first important work to be done in the testing of the plant is to determine whether or not the various connections are tight. The water, steam, air or brine piping should be tested first, after which the ammonia connections should be tested thoroughly.

Testing With Air Pressure.—As previously indicated, in order to test the various connections for tightness, the system should be submitted to pressure test with air or carbon dioxide. In both cases care must be exercised to keep the pressure in a safe range. Ammonia systems may be tested with air but carbon dioxide gas is used for testing Freon or other low pressure systems. The following method is advisable for air pressure test: The water valves on the ammonia condenser and the compressor water-jacket on the compressor should be opened. The compressor should be started operating at about half of full-speed if it is possible taking air in on the suction side and discharging it into the system. The compressor should be operated during short intervals of time, stopping the compressor to allow the compressed air to cool. This method should be continued until the pressure shown by the high and low pressure gauges is 150 lbs. When this pressure has been reached, the main liquid valve at the ammonia receiver outlet and the stop valve at the low pressure gauge should be closed. The compressor should be again started, and the same method of procedure followed until the pressure in the high-pressure system is 275 to 300 lbs. per sq. in. gauge.

During the foregoing procedure, the operator should watch all parts of the apparatus carefully. This is especially true of the bearings. They should not be allowed to get any hotter than bearable to the hand, or a temperature of about 125° F. The system should be allowed to stand under pressure for a period of 12 to 24 hours. Taking into consideration the change of the temperature of the air about the apparatus, the loss of pressure during this time should not amount to more than 5 or 6 lbs. While the pressure is on the system, the system should be gone over carefully to discover any leaks. The larger leaks may be discovered by listening along the various connections. In order to make sure that there are no small leaks, the various connections should be covered with soap-suds, the leakage of air causing the formation of bubbles. The condenser and ammonia receiver should be blown out by opening the receiver drain valve. This allows particles of dust, dirt and scale to be blown from the apparatus.

In a similar manner, the evaporating coils may be blown out by pumping high pressure on the high pressure system and then blowing the coils out with this high pressure air through the expansion valve by disconnecting the refrigerating coils or expansion coils at the ends. After the system has been tested with air pressure in this manner, it is advisable to inspect the compressor valves to make sure that scale and other foreign matter are not lodged in these. After this, the whole system can be retested by pumping a pressure of 150 lbs. on the entire system. The air pressure may be released by opening the valve on the scale trap.

Testing Under Vacuum.—It is usually advantageous to test the plant under a vacuum. The compressor should be operated to draw air from the system and discharge it to the atmosphere until the high and low pressure gauges register a fairly high vacuum, 26 to 28 in. of mercury. There should be no appreciable rise of the pressure after a considerable length of time of ten to eighteen hours.

Charging Anhydrous Ammonia.—For removing the ammonia from a cylinder, the iron valve in one end of each drum is so arranged that it has a bent pipe inside the cylinder which reaches nearly to the wall of the drum. By tilting the cylinder so that the valve end is three or four inches lower than the opposite end, and so placing the cylinder that this bent pipe faces down, which is indicated by the valve opening facing up, and also by the brass tag which will be found on the valve end of the cylinder on the upper side, the cylinder is then in proper position for withdrawing ammonia. When ammonia is put into the cylinder the position of the cylinder should be reversed so that the pipe extends and faces upward. These positions are indicated by Figs. 184 and 185 respectively.

In charging anhydrous ammonia into the refrigerating system, the benefit of the evaporation of the liquid anhydrous which is charged may be secured in the evaporator. The pipe connection should preferably go from the ammonia drum to a tee in the pipe on the low pressure side of the expansion valve as shown by Fig. 184. Permanent connection to the system by means of a tee and valve at this point should exist on every system unless, as in Fig. 186 showing the receiver, the system is provided with a three-way valve for the main liquid valve, in which case the proper outlet of this valve may be used for a charging connection. It is recommended by some engineers to have a check valve in the line from the ammonia drum between the charging valve on the system and the valve on the ammonia drum, as indicated by Fig. 184. This check valve will prevent any return of liquid anhydrous ammonia back into the drum due to a partial vacuum being formed in it.

In charging a new compression system, after it has been properly tested for tightness by air pressure, the compressor is started, the system being pumped down to a vacuum; the compressor is then stopped, and the expansion valve on the liquid receiver closed. The anhydrous ammonia is slowly admitted through the connection from the ammonia drum, which is properly placed and tilted so that the bent pipe inside the drum extends to the very lowest part. The water is turned into the condenser. The anhydrous ammonia should be admitted slowly and as soon as the gauge on the suction side begins to show pressure, start up the compressor and run slowly. After the

pressure as shown on the suction gauge has reached about 15 lbs., the compressor should be speeded up to run at normal speed. The charging valve should be opened sufficiently to take in ammonia from the drum to maintain this pressure. Charging from ammonia drums should be continued until sufficient anhydrous ammonia has been obtained in the receiver to thoroughly seal the liquid line to the expansion valve and the receiver is one-half to two-thirds full.

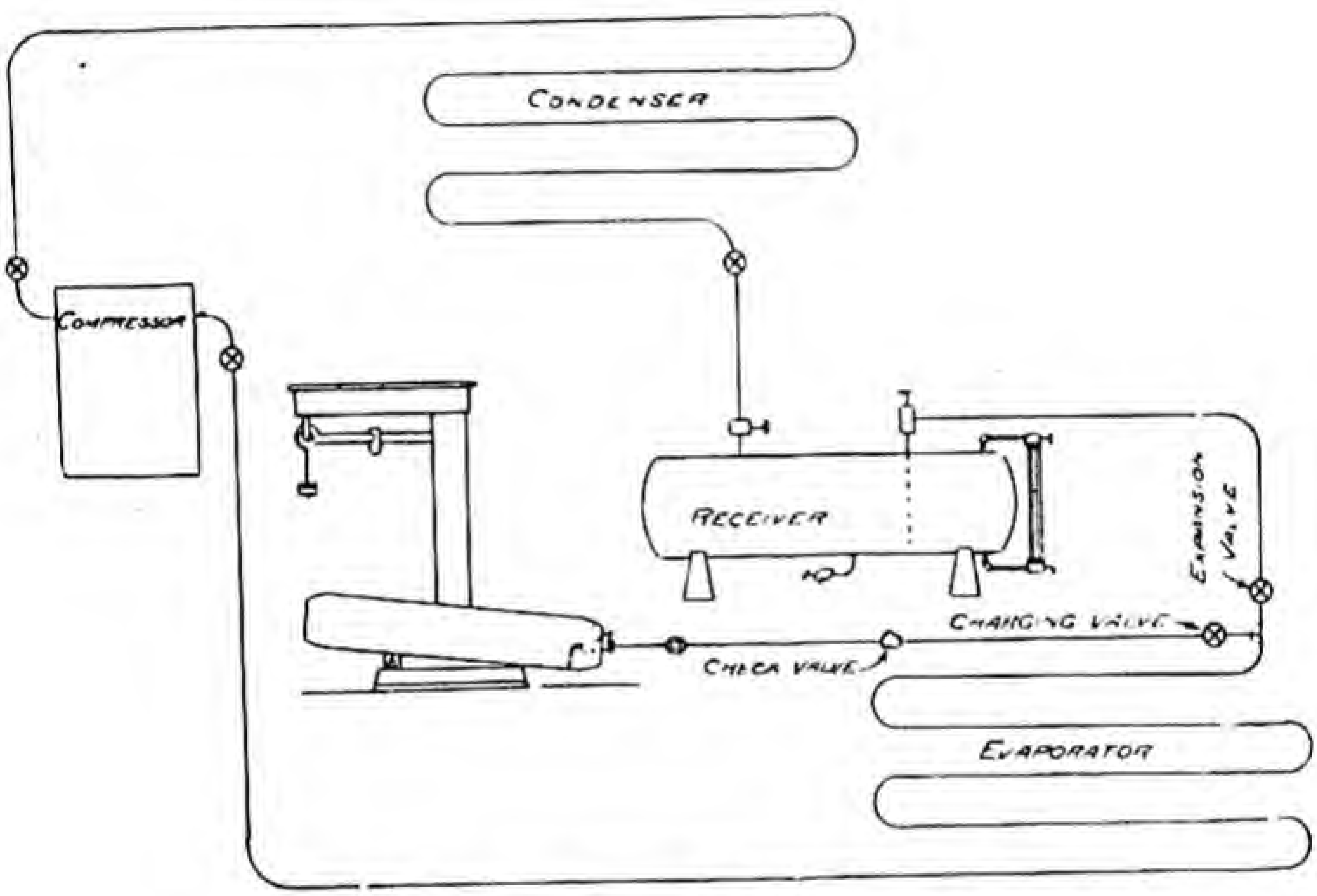

Fig. 184.—Connections for Charging System.

All receivers for anhydrous ammonia should be equipped with gauge glasses to indicate the level of ammonia contained in them. Such gauge glasses are made with automatic ball valves so that if the glass is broken the ball immediately closes the valve and no ammonia can escape. Without such a gauge glass on the receiver it is more difficult to operate the refrigerating system and more skill and experience is required to tell when a proper amount of ammonia is in the system or whether ammonia is running low and the ammonia going to the expansion valve is all liquid or part gas.

The pressure of the ammonia in the drum will generally force the liquid ammonia out through the bent pipe and charging connections into the suction side of the refrigerating system. However, it should be observed that the pressure in the drum will correspond to the temperature of the saturated ammonia when the drum valve is being used as an expansion valve. The pressures corresponding to the saturated temperatures of ammonia are shown by the following table:

Temperatures of ammonia, deg. F.	Corresponding pressure lbs. per sq. in. gauge
0	15.7
20	33.5
40	58.6
60	92.9
80	138.3
100	197.2

The foregoing table shows that the temperature of the ammonia in the drum should be taken into consideration when attempting to charge liquid into the system. For example, if the drum is taken in from the outside on a cold winter day when the temperature of the liquid ammonia in the drum may be 0° F., it will be difficult to charge

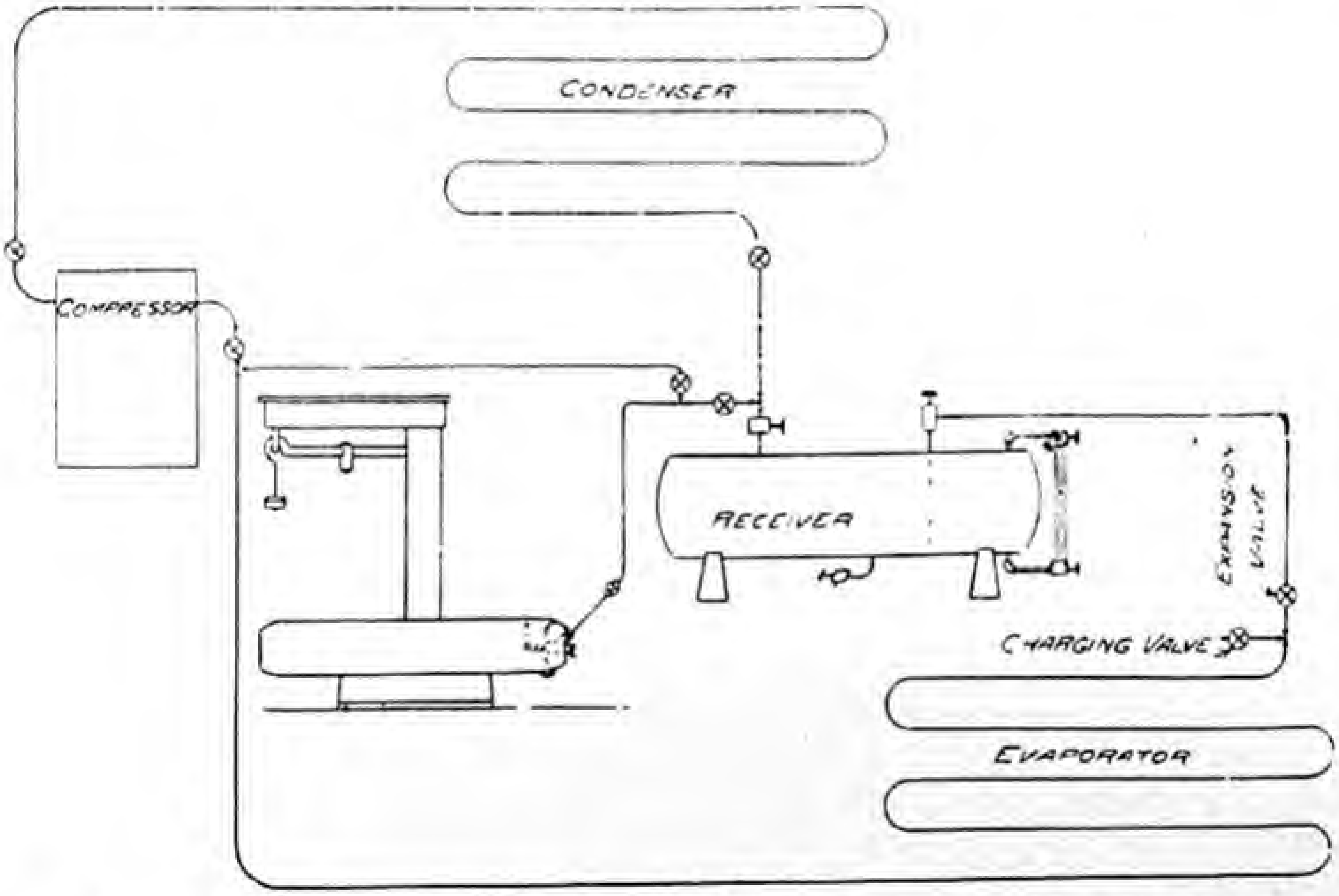

Fig. 185.—Connections for Removing Refrigerant from System.

ammonia into a refrigerating system having a pressure on the low pressure side of 20 lbs. per sq. in. gauge. In this case, it would be necessary to reduce the pressure on the low pressure side, or wait until the pressure of the ammonia has raised due to the increase of temperature.

When the last of the liquid passes out of the drum, the operator listening for an indication of this will notice the beginning of a hissing or whistling sound in the liquid pipe. The valve should then be closed. Some engineers wait until frost appears on the lower end of the drum and then will close the valve. Others wait until this frost which appears has melted, which will indicate that all the liquid ammonia has passed out of the cylinder. When the drum is warm this is a fairly

dependable sign. The sound of the cylinder caused by tapping it is also an indication when the cylinder becomes empty. Not infrequently, cylinders are shipped back to the manufacturers containing considerable liquid ammonia. Ammonia cylinders should be weighed on the scales after disconnecting, and the weight compared with the tare weight which is stamped on the brass tag attached to the drum. Should scales for weighing the drums not be available, the drums should be tested to see if any ammonia remains after being disconnected, as follows: Place the cylinder practically in the same position as it occupied when connected, and slightly "crack" the valve. If any liquid ammonia remains in the cylinder it will then be forced out by the pressure of vaporizing ammonia. This will show the cylinder is not empty, in which event it can be reconnected and entirely emptied. If only gas comes out of the cylinder in this position, the cylinder can be assumed to contain no liquid and it will not be necessary to reconnect it.

Amount of Ammonia.—Ordinarily about 25 or 30 lbs. of anhydrous ammonia per ton of refrigerating capacity is considered the proper charge for a plant. Ice making tanks operating on the dry expansion system will require approximately 50 lbs. per ton of ice. Brine cooler ice tanks will require approximately 70 lbs. per ton of ice. This may vary somewhat, according to the type of plant. With considerable direct-expansion piping for cold storage and business houses and the like, 1/3-lb. to 1/5-lb. of ammonia per lin. ft. of 2-in. direct-expansion piping, or 1/5-lb. to 1/7-lb. per lin. ft. of 1¼-in. piping are the quantities often used in figuring the proper amount of ammonia for the system. With a flooded system, more ammonia than this is, of course, required.

Anhydrous ammonia is very hard to retain in the system, and through a very small leak in the course of time a considerable amount may escape. With a deficiency in the charge of ammonia, a plant will not operate properly. If after a period of operation the condenser runs excessively hot, and very little liquid comes down to the receiver, pipe lines will not frost well and the machine does not perform its work properly, this is an indication that the ammonia charge is low. Often the ammonia in the receiver gets below the end of the outlet pipe to the expansion valve. When this occurs, hot gas from the condenser, as well as liquid ammonia, passes along to the expansion valve. This condition can only be remedied and the useless work thus being done eliminated by charging more ammonia into the system. When there are no gauge glasses on the receiver to indicate when the liquid level is lower than the outlet pipe, this may be detected by listening to the sound of flowing liquid in the main liquid line to the expansion valve. When any gas is flowing along with the liquid, a whistling or hissing

sound will be made. This may also be heard at the receiver. When this liquid pipe is not sealed and considerable hot gas flows through from the condenser, the gauge pressures will also not be normal, the condensing pressure being lower and the low pressure being greater.

In replenishing the ammonia charge of a compression plant, the connections of the shipping drum to the system should be made as previously described, and indicated in Figs. 184 and 186. The location of the ammonia drum should be the same, and the method of charging previously described be followed out, except that it is not necessary to pump the system down to a vacuum.

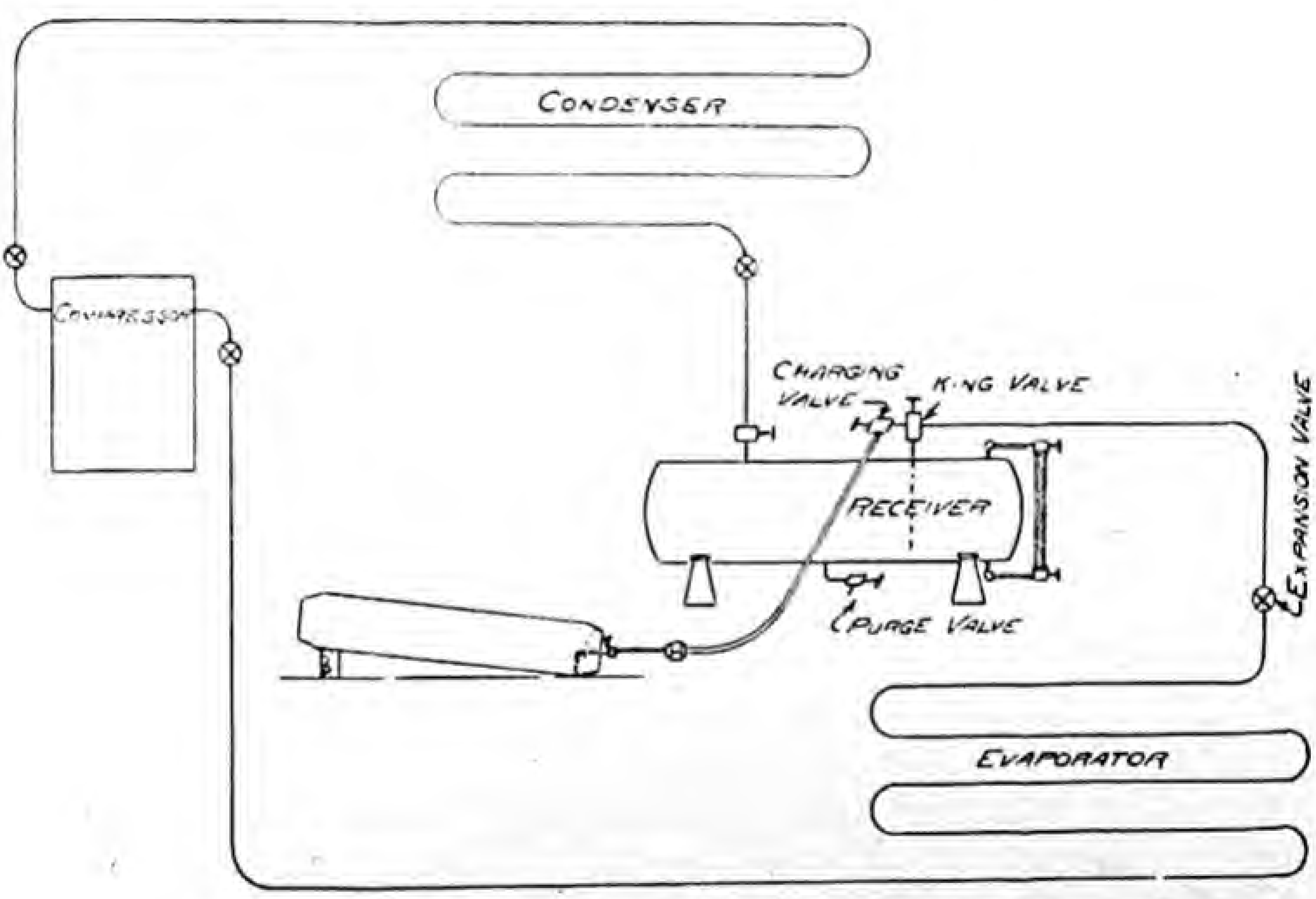

Fig. 186.—Connections for Charging System.

It is an easy matter to overcharge a plant, and when overcharged it is difficult to operate the plant satisfactorily. The temperatures and pressures cannot be controlled as well as when the plant contains a normal charge. There is also danger of explosion when the plant is overcharged, particularly if it becomes necessary to pump out any part of the apparatus and store the ammonia in another part, as in the case of storing all the ammonia in the condenser and receiver. When this is done, as it is often in winter time when the plant is closed down for repairing and overhauling, the valve between the condenser and receiver should not be closed since any change in temperature would cause expansion and create a pressure in the part where the ammonia was enclosed. Always charge a plant in small quantities until the plant operates normally.

As long as frost appears on the suction connections up to a point near the compressor, there is sufficient ammonia in the system when the plant is operating under the usual working conditions. Under this condition, there should be considerable liquid ammonia in the receiver. If the ammonia is low in the liquid receiver and the frost does not come back on the suction connections as it should, this is generally an indication of a lack of ammonia.

After the ammonia system has been charged in this manner, the various connections, joints, etc., should be tested for ammonia leaks.

Ammonia Leaks.—Ammonia leaks in the atmosphere are first detected by the odor of ammonia, and then may be located by so-called test paper which, upon being moistened and coming in contact with ammonia fumes turns to a pink color; or they may be located by burning sulphur sticks, the gas from which coming in contact with escaping ammonia will produce a white cloud or smoke. Leaks in brine tanks or into condensing water are not so easily detected. The presence of ammonia in brine or water may be detected by the same test paper which, when moistened, will detect leaks in the air. Ammonia in water or brine may also be detected by Nessler's solution.

Phenolphthalein Paper.—The customary and most convenient test for leaks of ammonia is the moistened phenolphthalein paper, which turns pink. Phenolphthalein is a crystalline white substance which alkalies turn red or pink, but which acids will again decolorize. This is the paper which is commonly supplied by ammonia manufacturers. It can be made in the following manner:

Dissolve one gram (15½ grains) of phenolphthalein in one litre of weak alcohol. Then immerse in this strips of unsized paper, and hang up to dry, after which it is ready for use. Soft chemical filter paper is a good paper to use for this purpose.

This test paper which turns red has been sometimes erroneously called litmus paper. Litmus paper, however, in its neutral state has a faint blue or lavender color, and when exposed in a moist state to ammonia or other alkalies turns *blue* and will turn *red* when exposed to an acid. Litmus paper is perhaps not as good for detecting ammonia leaks as phenolphthalein paper, since it requires more practice to tell the neutral color of litmus. The slightest colorating of pink in phenolphthalein paper indicates the presence of ammonia.

Sulphur Sticks.—Sulphur sticks may be prepared in the following manner: Take two or three ounces of pulverized or lump sulphur, powdered up, and heat in a tin cup over a candle or gas jet until the sulphur becomes liquid. Then dip some pine splints, about 1 ft. long and ¼-in. square into this sulphur and allow the sulphur to harden. Care should

be observed that the sulphur is not set on fire by the high temperature of the flame.

One of these sulphur sticks may be lighted with a match or from the flame of a candle and passed slowly around the ammonia pipe where it is suspected a leak exists. The fumes of the sulphur uniting with the ammonia will produce a heavy white smoke, which will indicate the location of the leak.

Nessler's Solution.—Nessler's solution is a more sensitive indicator of traces of ammonia in solution than is the phenolphthalein paper. This solution is used by adding a few drops of it to water or salt brine which is to be tested. If any ammonia is present in the water or salt brine, a yellow coloration of the liquid will take place, which turns to a full brown when a quantity of ammonia is present. The degree of coloration is an index as to the approximate amount of ammonia present. When this reagent is to be used to detect ammonia in calcium chloride brine the brine must be treated as follows: Dilute one ounce of the calcium brine with four ounces of water and add a sufficient quantity of concentrated solution of pure sodium carbonate to precipitate all the calcium in solution. Then filter and add to the solution a few drops of Nessler's reagent. If the solution turns brown, ammonia was present in the original calcium chloride brine.

Nessler's solution may be made in the following manner: Four and a half drams (apothecaries') mercuric chloride are dissolved in 10 fluid ounces of water and 1 ounce and 1 dram of potassium iodide are dissolved in 3½ fluid ounces of water; add the former to the latter until a slight permanent precipitate is produced. Next dissolve 3 ounces and 7 drams of potassium hydrate (caustic potash), in 7 fluid ounces of water, let it cool and add same to the foregoing solution, and then dilute the whole with distilled water until it makes one quart.

The metric formula for making Nessler's solution is as follows: Dissolve 17 grams of mercuric chloride in about 300 c.c. of distilled water; dissolve 35 grams of potassium iodide in 100 c.c. of water; add the former solution to the latter, with constant stirring, until a slight permanent red precipitate is produced. Next dissolve 120 grams of potassium hydrate in about 200 c.c. of water; allow the solution to cool; add it to the above solution, and make up with water to one litre; then add mercuric chloride solution until a permanent precipitate again forms; allow to stand until settled, and decant off the clear solution for use; keep it in glass stoppered blue bottles and set away in a dark place to keep it from decomposing.

Miscellaneous Operations.—If it is desired to pump the ammonia out of any part of the low pressure system, this may be done by proceeding as follows: The compressor should be operating as usual and

if the entire low pressure side is to be pumped out the main liquid valve on the outlet of the receiver should be closed. The operation should be continued until the low pressure gauge registers about zero pounds. At this time the compressor should be stopped and after waiting about one hour the compressor should be started again and continued in operation until the vacuum is increased to about 25 in. of mercury.

If individual parts of the low pressure side are to be pumped out, it is necessary to close all of the valves on these parts except the ones which are to be pumped out. The expansion valve to this part should be closed and the operation continued as above indicated. If it is desired to pump the ammonia out of any section of the ammonia condenser, the machine should be operated as usual, closing the gas inlet and the liquid outlet valve on the section of the condenser to be pumped out, and opening the pump-out valve on this section.

In pumping out in this manner the operating engineer will have to use his judgment as to the extent of the removal of ammonia. This in general is indicated by the frost on the pumpout line and if possible it is desirable, after first reducing the pressure to a vacuum, to stop the pumping out process for one or two hours to allow the liquid in the condenser to evaporate. After this, the above process should be repeated, pumping down the section to the desired vacuum. Of course, the water should be drained from the condenser in order to prevent freezing.

Purging of Ammonia Condensers.—The presence of non-condensable gases in the condenser accounts for the unusually high pressure in most cases. Ordinarily, the pressure of the ammonia will be such that the temperature of the ammonia in the liquefying portion will be 5° to 10° F. above the temperature of the water leaving the condenser. The temperature of the water off the condenser may be determined, and by adding 5° to 10° F. to this, the temperature of the saturated ammonia may be obtained. Then by reference to a table of the properties of saturated ammonia the corresponding pressure may be ascertained. The operator can then compare this with the actual work pressure and note the difference. The condenser pressure that should be obtained under ordinary conditions while the plant is in operation is shown by the following table with various initial temperatures of the condenser water:

Initial temp. of condenser water, deg. F.	Condenser gauge pressure lbs. per sq. in.
50	120
60	140
70	160
80	180
90	200

The pressures indicated in this table are considered good practice. If the pressures are materially above those indicated in the foregoing table for the various initial temperatures of condenser water, the operator should suspect the presence of non-condensable gases in the condenser and should proceed to remove them.

It is sometimes advantageous to note the reading of the high pressure gauge when the machine is stopped and the water is flowing over the condenser at full force. These pressures are shown by the following table when the non-condensable gases are present in small quantities only:

Temp. of air and water over condenser	Condenser pressure lbs. per sq. in. gauge
50	74.5
60	92.9
70	114.1
80	138.3
90	165.9
100	197.2

The foregoing table shows the pressures which correspond to the saturated temperatures when the ammonia is pure. It is evident that any non-condensable gas will increase the total pressure in proportion to the partial pressure of the non-condensable gas.

Thus if the pressures registered by the high pressure gauge are considerably higher than those shown by the table it is an indication of the presence of non-condensable gases and that the condenser should be purged.

The purging that is required varies considerably with the type and size of plant. In some plants it may be necessary to purge the condenser weekly while others would need to be purged only every month or so. It is apparent that an excessive amount of purging causes a waste of ammonia and should be resorted to only when the pressures demand it. In order to remove the non-condensable gases the coils should be allowed to become as nearly full as possible with liquid and then should be cut out of service by the manipulation of the proper valves.

The condensing water should be allowed to flow for two to five hours over the sections to be purged to allow the ammonia to condense and the gases to separate from the liquid ammonia as much as possible. The purge valve on each section or at the top of the condenser should be connected to a pail or barrel of water if it is desirable to eliminate the disagreeable odor of ammonia. The purge valve should then be opened only slightly. As long as the permanent gases are passing into the water it will simply bubble. But as soon as ammonia comes into the water the familiar cracking sound is produced which is similar to the sound of steam being introduced into water and the purge valve

should be closed immediately. After an interval of a few minutes the process should be repeated. Usually a few purgings in this manner will remove all of the non-condensable gases from the condenser.

Withdrawing Ammonia.—Ammonia should not be withdrawn from a plant and placed in shipping drums without weighing the quantity of ammonia placed in each drum. The danger due to overfilling of these drums is very great. The maximum weight which should be charged into a given sized cylinder is shown in the following table:

Maximum weight	Size of drums
150	12 in. in diameter by 7 ft. 0 in. long
100	10 in. in diameter by 7 ft. 0 in. long
50	10 in. in diameter by 3 ft. 10 in. long
40	10 in. in diameter by 3 ft. 6 in. long
25	10 in. in diameter by 2 ft. 4 in. long

The empty cylinders should be placed on scales in a horizontal position with the valve stem pointing downward, or so placed that the bent pipe inside of the cylinder extends to the top, thus permitting the cylinder to be nearly filled. The cylinder should be connected with a tee, one line from which goes to the receiver and the other side of the tee is connected to the suction side of the machine. There should be a valve in both lines. Upon opening the valve from the receiver a small quantity of liquid ammonia will run into the cylinder, and then if this valve is closed and the valve to the suction line opened, the ammonia which ran into the drum will evaporate, carrying off air to the suction line and reducing the temperature in the drum. Upon closing the valve to the suction line and again opening the receiver valve liquid ammonia will run freely into the shipping cylinder. The scale beam will show when a safe quantity of liquid has been drawn into the drum. In case it is not convenient to connect one side of the tee to the suction side, this end of the tee may have a valve attached so that the ammonia which otherwise would go to the suction side of the machine may be allowed to escape. This is only a small quantity of ammonia and will represent but a slight loss. The proper connections and positions of the shipping drum are shown in Fig. 185.

Minor Operations.—Several other points about the plant need to receive attention during the period that the apparatus is first put into operation. It is evident that some of the oil which gets into the compressor through the piston rod stuffing box will be vaporized and therefore pass over into the oil interceptor and condenser. The function of the oil interceptor is to intercept as much as possible the oil as it comes from the compressor. Thus the oil interceptor should be drained at suitable intervals. In a similar manner the suction strainer should receive attention. This is especially true of the first few days

of operation of the plant. Scale or other foreign matter may collect to a considerable amount at this point. If the strainer is not cleaned, it may become clogged, under which condition the pressure in the ammonia compressor may be considerably lower than the pressure in the suction main. The detrimental effects of this condition are apparent. The stop valves on the pressure gauge line should also be given attention. These should be so regulated that the movement of the hands on the gauges will not be less than ⅛-in. at the end of the pointer and not more than ¼-in. These valves should be opened wide every morning while the compressor is in operation for a few seconds after which they should be nearly closed. By this means the operator may make sure that the gauges are indicating the proper pressures.

QUESTIONS ON CHAPTERS XVII.

1. What are some general factors to be considered in selecting a suitable location of the refrigerating plant, and in the arranging of the apparatus in it?

2. Describe the method of making foundations for refrigerating machines.

3. Describe the general methods employed in lining up foundations, templets, and machinery.

4. Describe three methods of making pipe connections for ammonia work.

5. What things should be done in installing refrigerating equipment before the machinery is started?

6. Describe the precautions to be taken when the equipment is being tested under air pressure.

7. Describe the various methods of charging ammonia into the refrigerating system.

8. Describe the method for detecting the presence of ammonia in air.

9. How is the presence of ammonia detected in liquids such as water and brine?

10. When and how should ammonia condensers be purged?

CHAPTER XVIII.

OPERATION AND CARE OF APPARATUS.*

Operation of Mechanical Equipment.—In the operation of the mechanical equipment for a refrigerating or ice making plant the operating engineer should remember that it is his primary duty to get a certain desired result with a given apparatus. He should further remember that his duty is not only the accomplishment of a desired result, but that he should produce the maximum of result with the minimum amount of expense.

The various parts of the apparatus must be kept in such a condition that they will function properly, not only within themselves, but in union with other parts of the mechanical equipment. The prime mover for a refrigerating machine must be kept in such condition that it will operate the machine at the desired speed. The condensing surfaces of the condensers must be kept in such condition that the condenser pressure will not become too high. The temperature of the refrigerant must be maintained at such a point as to produce the desired temperature in the ice making tank or the cold storage room, and, as previously indicated, it should be remembered that the suction pressure should not be carried lower than is necessary for the results desired.

From this it will be observed that the operating engineer is not only charged with the daily mechanical manipulation of the equipment, but it is also his duty to keep the equipment in as nearly perfect mechanical condition as possible. To do this the engineer must inspect the apparatus daily and overhaul and repair it so that it is always in a serviceable condition.

Importance of Overhauling and Repairing.—The maintaining of all parts of the apparatus in a serviceable condition is a consideration of great economic importance in the life of apparatus. It not only lengthens the useful life, but also allows the plant to produce the required work in the most efficient manner.

* Considerable matter in this chapter has appeared in *Ice and Refrigeration* and *Power* at various times, and permission has been granted for its re-use here.

The existence of such a condition is mutually beneficial to the engineer and the plant owner. If the plant is operated in an efficient manner the owner will be able to make a reasonable profit on his investment in the plant. On the other hand the operating engineer will be benefited in a number of ways. He will be compensated in a financial manner in proportion to the magnitude of the profits which he is able to help make for his employer. Also he will receive the recognition and prestige of an expert operating engineer.

Furthermore, the importance of the proper overhauling and repairing or the need of renovation of the plant in general, may be more fully appreciated when one considers the great losses that might occur due to stoppage of the refrigeration plant. The large cold storage warehouse may be considered as an example. These warehouses are filled mostly with perishable food products. It is easy to imagine the loss that would occur should the refrigeration service fail, due to a breakdown on account of inadequate repairs. It is obvious that all parts of the plant must be kept in good repair in order to maintain a dependable and reliable refrigeration service.

Depreciation of the Apparatus.—Depreciation is the loss of value. This may be caused by the loss of the useful life of the plant unit or its parts; or it may be caused by the invention or the design of a more efficient apparatus; also, it may be brought about by the apparatus becoming too small for the economical operation of the plant; and it may be occasioned by a drop of prices of apparatus. From the above it is evident that depreciation may be classified under two general heads; they are natural depreciation and functional depreciation.

Natural Depreciation.—Natural depreciation is loss of value due to physical and chemical changes in the apparatus. Rusting of ice cans and condenser coils is an example of chemical action; electrolysis of the metals of the brine circulating system is an example of chemical and electrical action; abrasion or wear is an example of physical or mechanical action; rotting of wood and insulation is an example of chemical action. Loss of value due to accident may also be considered natural depreciation.

By making adequate repairs of all defects as soon as conveniently possible the engineer is able to retard the natural depreciation. Deterioration of the apparatus is hastened if it is kept in service without repairing the defects. This is true of all equipment, such as compressors, condensers, coils, insulation. Unless the defects are repaired they will increase and may lead to expensive repairs, renewals, or even the loss of property and human life.

Functional Depreciation.—Functional depreciation is the loss of value due to obsolescence, inadequacy, or drop of prices. Obsolescence

is brought about by improvements in the construction and design of apparatus. Inadequacy arises from increased demands upon the plant apparatus until the plant units are too small for economical operation under the increased load. Drop in prices is caused by anything that increases the supply relative to the demand. It is evident that these factors are not within the immediate province of the operating engineer. However, it is obvious that he may retard the rapid natural depreciation of the apparatus.

In the ultimate analysis it will be noted that considerations of depreciation of apparatus lead to the question of determining when it is desirable to retire or scrap the old apparatus. It is evident that there will be a time when the apparatus may be considered worn out and out of date.

Renewal of Apparatus.—It is a very unsuccessful policy to try to repair the worn apparatus or parts thereof too often. Generally the total cost in the end will be more than that of a new apparatus or part thereof. There should be no hesitancy about the purchase of new apparatus to take the place of the worn-out and out-of-date equipment. Generally the management adopts the policy of limiting the funds for repairs and renewals. It should be entirely obvious that this course of procedure leads only to inefficiency. The operating engineer should call to the attention of the management the amount of apparatus and machinery which is necessary for all the improvement and indicate to the management thoroughly that the improvements are essential to successful operation of the plant.

Hence, the proposition of the renewal of the apparatus is a consideration having great economic importance. The success of a business enterprise often depends upon the judgment used in scrapping the older apparatus to make way for the newer apparatus. Generally speaking, it may be stated that it is profitable to retire the old apparatus when the yearly profit can be increased by installing the more economical apparatus. In determining the yearly cost of operation the natural depreciation should be classed with repairs, while the functional depreciation should be offset by an annuity for replacement of the apparatus at the expiration of its useful life.

The desirability of retiring the old apparatus or plant may be determined by noting the total annual costs of operation of the old and the new or proposed apparatus. The annual cost of operation of the new apparatus is made up of such items as follows: Operating expense, repairs and taxes, functional depreciation and interest on the investment. The annual cost of operation of the old unit at the time of renewal would be made up of the following: Operating expense.

repairs and taxes and the interest on the salvage value of the equipment. Knowing the annual cost, the annual profit may be found by subtracting the annual cost from the annual income.

Thus, the question of renewals of apparatus, machinery, etc., resolves itself into a consideration of the dollars and cents of profit or loss at the end of the year.

Routine of Overhauling.—In order to live up to the standards of his profession the operating engineer should be systematic and accurate in his overhauling and repairing work. During that part of the year previous to the overhauling season he should note in a systematic manner such things as repairs which are necessary, general improvements, desirability of retiring old apparatus and new installations of machinery. The daily engine room log will enable him to study the efficiency of the plant during the year. Hence, at the beginning of the overhauling season, he will have a definite and systematic plan of action.

The engineer should begin his work of overhauling the plant by repairing all defects in the apparatus and machinery; he should test and inspect all parts of the plant thoroughly; he should then clean and paint all parts of the plant that require cleaning and painting. Hence, by much care and attention, the engineer may put his plant into perfect adjustment and maintain the apparatus and machinery in first-class mechanical condition.

In order to proceed with the work of overhauling and repairing the various parts of the refrigeration apparatus in an efficient manner the engineer should outline the actual work in a detailed manner, so that everything receives attention and that nothing is overlooked. Some of the parts requiring attention are shown by the following outline:

1. *Compressor.*—Suction and discharge valves, cylinder walls, piston-rings, stuffing-box and packing, piston-rod, bearings, joints.
2. *Oil Interceptors.*—Drain valve interior surface, joints.
3. *Condensers.*—Valves, fittings, joints, outside of pipes, inside of pipes.
4. *Receivers.*—Valves, glass gauges, drain valve.
5. *Ammonia Connections.*—Valves, joints, supports, insulation, paint.
6. *Direct Expansion Piping and Brine Piping.*—Valves, joints, pipes, fittings, inside of pipes, outside of pipes.
7. *Brine Coolers.*—Valves, joints, fittings, gauge columns, inside, outside, insulation.
8. *Brine Circulation System.*—Pump bearings, valves and stuffing boxes, valves, fittings, joints, connections, insulation.
9. *Ice Freezing Tanks.*—Woodwork, steelwork, insulation, cans, agitators, hoisting equipment, thawing and dumping equipment.
10. *Raw Water Equipment.*—Softeners, filters, storage tanks, air blowers and compressors, valves, fittings and connections.
11. *Distilled Water Apparatus.*—Separators, steam condenser, reboiler, flat cooler, filter, storage tanks, connections, valves, fittings.
12. *Ammonia.*—Quantity and purity.
13. *Brine.*—Quantity and purity.

With due consideration to the foregoing facts, it is apparent that the overhauling and repairing the parts of the refrigeration plant in a proper manner requires considerable thought and work. Therefore, the operating engineer should begin this work early in the overhauling season, so that he may finish his work in a satisfactory manner.

Overhauling the Compressor.—Since the compressor is probably the most important part of the compression system, considerations of the careful inspection and thorough examination of same will be given attention first. Also, since the compressor contains a majority of the reciprocating parts, defects are liable to occur sooner in the compressor than in the other parts of the system. The action of the valves, packing, piston-rod, crosshead, connecting rod and crankshaft should be observed closely while the machine is in operation, so that any irregularities may be noted. Indicating the compressor will help very much in ascertaining the operation of the valves and piston-rings.

After having observed carefully the functioning of these parts the compressor may be shut down for overhauling.

In the main the work of overhauling the compressor consists of examining thoroughly and putting into serviceable condition each and every part which may be subjected to wear. To do this in a proper manner the machine must be taken apart. This facilitates the inspection and examination.

Valves.—The suction and discharge valves are important parts in the efficient operation of the compressor. The valves should be taken apart and examined closely. Generally they should be ground to a perfect seat and the springs should be adjusted for the quiet operation of the valves. If the seat between the valve cage and the cylinder head is not perfect this also should be ground to a perfect seat.

Packing, Piston-Rod and Piston.—The packing, piston-rod and piston should be removed and subjected to careful inspection. The piston-rings should receive especial attention. If these are only slightly worn they may be peen-hammered to force them out against the cylinder walls.

If the rings are worn considerably and are loose in the grooves, or show any other defects, they should be replaced by new rings. The cylinder walls should be examined closely. If the surface has a high metallic polish this is an indication that the cylinder is properly lubricated and that the piston-rings are in good condition. If the surface has a dull appearance with traces of wear or abrasion the cause of this condition should be removed at once. Likewise, should the cylinder be out-of-round, it should be rebored and refinished.

In a like manner the piston-rod should be examined for wear and shoulders. In the event that the rod is worn it should be replaced or

a new set of good fibrous packing should be installed. In the event that the rod is out of line, due to the wear of the piston on the cylinder, it may be advisable to pack the rod with eccentric fibrous packing rings in order to compensate for the misalignment. If the rod is true and is well aligned a good metallic packing may be used. In the event that the rod is worn only slightly it may be turned down and refinished. A new set of packing to fit the rod should then be installed. However, it is evident that the operating engineer is responsible for the daily performance of the packing and rod, and it is his duty to give these important parts of the compressor their proper care, adjustment and lubrication.

Bearings.—The bearings between the frame and the crosshead should be examined. The crosshead shoes may need rebabbitting. The shoes should be adjusted to compensate for the wear on the frame and shoes.

The crankshaft main bearings and the crankpin and crosshead bearings should be inspected. In the event that these are worn considerably they should be rebabbitted. If they are only slightly worn they should be made smooth and the oil grooves should be cut out in the proper manner.

Keys and wedges should be replaced or put in such a condition so that they may be used for adjustment during the coming working season. The crankpin and crosshead pin should be made true and adjusted properly to their bearings.

Before assembling the machine it is advisable to check the alignment of the frame with reference to the crankshaft and the prime mover. The machine must be well aligned for the proper operation of the bearings.

In reassembling the machine all parts should be put into proper adjustment; new gaskets should be used; the clearance in the cylinder should be adjusted to the minimum and for quiet operation when the machine is warm. Indicators and pressure gauges will facilitate the proper adjustment of the suction and discharge valves.

After the machine is reassembled it should be thoroughly cleaned and painted, if necessary.

Oil Interceptors and Traps.—All oil interceptors and separators on the high pressure side should be taken apart and cleaned, both inside and outside. All gaskets should be renewed if they are not in first-class condition. Interceptor drain valves should be inspected and examined. In general these valves should be fairly large in order to reduce the probability of becoming clogged with foreign matter.

Similarly the suction traps and strainers on the low pressure side should be examined and cleaned. Particular attention should be given

to the proper cleaning of these traps. Frozen oil, scale and other foreign matter may accumulate to the extent of partially closing the suction lines.

Ammonia Condensers.—Next to the compressor, the condenser, is probably the most important part of the refrigeration apparatus. Therefore the cleaning and keeping clean of the condenser surfaces are factors of importance. The function of the condenser surface is to transmit heat from the refrigerant to the water in order to cool and liquefy it. Hence, it is not logical to allow the pipes to become covered with a coating of foreign matter, which necessarily resists the flow of heat. A heavy coating of foreign matter requires a greater temperature difference between the refrigerant and the water in order to cause the necessary amount of heat transfer. The results in increased condenser pressure and hence makes the plant more expensive to operate.

Atmospheric Condensers. — The standard atmospheric ammonia condenser with its single gas connection at the top of the coil and its single liquid connection at the bottom is a type of condenser that is preferred by many operators. The arrangement of the connections to the coils is simple and practical. The condensing water is simply showered over the coils by the troughs. It has no small liquid drain connections that tend to rust away from the action of the water and the air. Leaks along the condenser pipes are easily discovered and readily stopped temporarily by a clamp and a rubber gasket. Leaks at the joints should be marked and repaired during the overhauling or shutdown period.

This type of condenser has a great advantage of being more nearly foolproof than any other type. However, it is being replaced somewhat by the newer designs of atmospheric condensers.

Water Distribution Over Atmospheric Condensers.—The securing of a proper water distribution over the individual condenser coils is a point of no small importance. The condenser is installed to conduct heat from the ammonia to the water; therefore it is essential to obtain an even distribution of water over all the pipes from end to end.

In actual operation proper care is not always given to leveling the water-distributing trough. The holes and slots in the trough tend to clog with algae, moss and other foreign material. These conditions affect the efficient distribution of the water. The galvanized troughs with serrated edges give the best service, everything being considered.

In order to determine whether the water is being distributed in the proper manner over each coil, the temperature of the water leaving the condenser coils should be ascertained. If there is a wide variation

of temperature of the leaving water it is on account of an uneven distribution of water to the sections. This is, of course, subjected to the assumption that the flow of ammonia to the coils is even and equally distributed to the various sections. To obtain the lowest possible condenser pressure the water should have the same range of temperature in passing over each coil.

Cleaning of Atmospheric Condensers.—The cleaning and the keeping clean of the condensing surfaces are of primary importance. The function of the surface is to conduct heat from the ammonia to the water. Therefore it is not logical to allow the pipes to become coated with a heavy layer of hard scale which will necessarily retard the flow of heat. Heavy scale increases the condenser pressure and thus makes the plant more expensive to operate.

As the water flows over the condenser there begins to form in the troughs and one of the upper pipes a sort of greenish slime. This gradually works downward and finally covers the entire condensing surface; it is deposited by nearly all kinds of water, but of course the amount will vary with the locality. If this foreign material is allowed to remain on the condenser it will harden into a scale. This begins at the top of the condenser, where the temperature is highest, and will gradually scale downward, as the efficiency of the higher pipes is being lowered by the accumulation of scale.

In the first place, it is evident that the scale should not be allowed to harden on the pipes. This may be prevented by washing the coils daily with a jet of water at high velocity. Rubbing the pipes with a stiff wire brush is a good practice.

In the second place, if hard scale is allowed to form it should be removed carefully and thoroughly. Scale may be removed from the pipes by means of a rasp or a chipping hammer. Chipping hammers should not be so sharp as to injure the pipe, for careless use of the cleaning tools may result in split pipes or injuries that will promote pitting and corrosion. After the coils have been thoroughly cleaned they should be covered with a good paint.

The inside surface of the condenser pipes should be kept as free as possible from oil and other matter. As oil is necessary for the lubrication of the compressor more or less of it finds its way over into the condenser. After the plant is shut down some of the condenser coils should be opened and examined for oil and foreign matter, after pumping out the coils, of course. To clean the coils the following procedure can be followed:

A steam connection is made to the gas header and then all gas valves are closed. The coils are disconnected at the bottom. Then, by turning steam into the gas header and opening the gas valves in

succession on each coil, the oil and foreign material are blown out of each section. The steam connection is removed and an air connection substituted. The coils are now blown out with high-pressure air to remove any condensed steam from the coils while they are still hot.

Valve seats and discs should be inspected. Gaskets for valve bonnets or housings, joints between valves and headers, and packing for valve stems should be renewed if the plant is to be operated continuously during the season. Other gaskets and packings should not be renewed unless it is certain they are not in first-class condition. If the bolts, studs or nuts are rusted considerably they should be replaced by new ones.

Operation of the Double-Pipe Ammonia Condenser.—The standard double-type ammonia condenser is similar to the standard atmospheric type in that it is as nearly foolproof as it is possible to make a condenser. There is no regulation of ammonia required. The entire performance of the various sections is controlled by observing the temperature of the water leaving the individual coils. These temperatures may be determined by inserting thermometers in suitable openings in return bends on tees or by feeling the last pipe at the end where the liquid ammonia leaves. The temperature should be approximately the same in order for the coil to operate evenly. The water outlet valves on each section may be adjusted so as to give the same outlet temperature of water on all of the coils.

Cleaning of Double-Pipe Condensers.—The removal of sediment and scale from the condenser surface is essential to the efficient operation of the condensers. The operator is under the disadvantage of not being able to see the condition of the inner tubes. Therefore he must wait for indications of an accumulation of scale or take active proceedings to insure that scale will not be allowed to form. When the condenser tubes become coated with sediment and scale the range of the temperature of the water in passing through the condenser decreases and the other indication of dirt is an increase of condenser pressure. The operator should not wait for increased condenser pressure but he should clean the surfaces regularly. Of course the frequency and exact method of cleaning the tubes depend upon the nature and amount of material deposited and the nature of the plant.

The sediment in the tubes should be prevented from forming a hard scale by removing it. This is usually accomplished by reversing the direction of the flow of water in the tubes for a few minutes each day or week, as the case may demand. This may be done by a proper connection of each section of the condenser to wash-out headers by means of three-way cocks and valves. The wash-out water is allowed to flow to the sewer.

In the event that sediment is allowed to form into hard scale it may be removed by a stiff wire brush or by a good tube scraper. This is fastened to a length of ½-in. or ¾-in. pipe and then pushed through the tubes to remove the scale. Thick and hard scale may be reamed from the tubes by fastening a drill upon the end of a length of ¾-in. pipe and then driving the drill through the tube with an air or electric motor, or by means of a pipe wrench. The tubes should be washed after cleaning in any of these ways.

Shell and Tube Condensers.—Both vertical and horizontal shell and tube condensers should be kept clean. If sludge or scale accumulate on the water side of the tubes they should be swabbed out at regular intervals during the operating season with a wire brush or cleaned with a power driven tube cleaner. The seasonal cleaning should remove all foreign matter from the tubes.

Many vertical shell and tube condensers have a space above the bottom tube sheet designed to accumulate oil settled out of the liquid refrigerant. This oil and any sediment present should be regularly drained out of the system. Whenever oil is being drained out of a system under pressure it should be remembered that the drain valve should not be opened wide. A larger proportion of the oil will be removed if the drain valve is set to allow the oil to pass out in a small stream just a little larger than is necessary to maintain flow. Otherwise the refrigerant will blow out with the oil while much of the latter remains in the vessel.

During the repairing and overhauling season the ammonia receiver or storage tank should be examined closely. The component parts of the valves and gauge glasses should be inspected thoroughly. The condition of the drain valve and connections should be noted. The drain valve should be fairly large in order to prevent clogging due to accumulation of scale, etc. All oil should be drained from the receiver.

Piping Systems.—The ammonia pipe lines which connect the various elements of the system should be carefully examined and repaired. The successful and smooth operation of the plant depends upon the design and layout of the piping system. Many difficulties of operation may be traced to neglect of the piping system.

The ammonia pipe lines with all joints and valves should be examined closely for leaks. A burning sulphur stick should be applied to all the joints and every foot of the pipe lines. If white fumes are given off by the burning sulphur, an ammonia leak is close at hand. After all leaks have been detected, they should be repaired promptly.

The failure of piping, valves, or fittings may be due to vibration and faulty methods of support and anchorage. In general, the lines should be level, well aligned and free from traps and pockets. Vibra-

tion in piping systems may be due to the intermittent flow of gas, vibration of the compressors and engines. Therefore it is necessary to make sure that all parts of the piping system are well supported and anchored.

The insulation of all cold pipes should be inspected and kept in first-class condition. The insulation may need to be renewed or repaired.

All the piping should be kept as accessible as possible. After all parts have been inspected and repaired, fresh paint should be used freely.

Direct Expansion Coils and Shell Coolers.—All joints, pipes, valves and fittings should be carefully tested for ammonia leaks in the manner previously mentioned. Any leak should be duly noted and repaired.

The condition of the inside and outside of the refrigerating pipes has an important bearing on the efficient operation of the plant. Refrigeration is produced by the evaporation of the liquid ammonia in the coils, the necessary heat being absorbed from the surroundings. The rate of production of refrigeration depends upon the amount of pipe surface, the mean temperature difference and the heat transfer rate of the evaporator wall. With the surface and temperature differences constant, it is evident that the refrigeration is in direct proportion to the transmission of heat by the wall of the evaporator.

The rate of this heat transmission depends upon the conductivity of the separating wall, the thickness of the wall and the surface effects. The accumulation of ice, snow, grease, oil, scale, or other foreign matter adds resistance to the flow of heat. Thus, the evaporating coils should be kept clean and free from oils in order to produce the most refrigeration with the least surface. Oil tends to collect in the evaporating coils and will remain there if definite procedure is not taken to remove it. Snow and ice tend to collect on the exterior surfaces of the evaporating coils in a similar manner.

In a shell brine cooler, the ammonia evaporates on the outside of the tubes and inside of the shell, and the brine flows through the inside of the pipes. This cooler tends to become coated with oil on the outside of the tubes, and ice may collect on the inside of the tubes in a manner quite similar to the evaporating coils.

The proper method of cleaning evaporated coils and shell coolers is to remove the ammonia and brine. Ice, scale, and other matter may be removed by mechanical means. In order to remove oil, dirt and other foreign matter from the coils and coolers, they should be blown out separately with steam, after which they should be thoroughly dried by blowing air through them while they are still hot.

All valves should be inspected, paying particular attention to the valve seats. Expansion valves should receive especial attention.

After being thoroughly cleaned inside and outside, the coils and coolers should be carefully reassembled. They should then be tested with air at 250 to 300 lbs. pressure. If the apparatus is tight and the atmosphere remains constant in temperature, the air pressure should fall only slightly in 24 hours. Any leaks in the coils or cooler may be detected by soapsuds or by submerging the apparatus in water.

Ammonia Tests.—With an absorption machine it is well to test the anhydrous ammonia at least once a day. If the dehydrator of the machine is working properly, the anhydrous ammonia should be perfectly dry, and if not it will, of course, contain a small percentage of water. The earlier absorption machines produced anhydrous ammonia having as much as one to two per cent of water, whereas present day machines with the efficient dehydrators used will produce very dry ammonia so that no water should be shown in the test for purity.

To tell how the absorption machine is operating, frequent tests should be made of the strong and weak aqua ammonia. This is best done by an apparatus attached permanently to the machine, which permits a sample of either the strong or weak liquor to be drawn into a receptacle in which floats a hydrometer. The whole is so enclosed that none of the ammonia vapors escaping from the liquor interfere with the operator.

The Beaumé hydrometer is the one most used. Tables giving the specific gravity, the per cent ammonia in solution, etc., are based on the Beaumé readings at 60° F. Hence, if the temperature at which the reading is taken is different than 60°, a correction for temperature should always be made as indicated on the correction table. Roughly, a difference of 15° F. is equal to a difference of one degree in strength. Thus, if the hydrometer indicates 25° Beaumé at a temperature of 45° F., this means the aqua would be 26° Beaumé if raised to a temperature of 60° F. (which is the standard temperature), while aqua at 75° F. indicating 27° on the hydrometer would be 26° aqua at standard temperature. Due allowance should also be made for the loss of gas in testing. Any sample of ammonia will lose gas rapidly when exposed to the atmosphere.

Purifying of Anhydrous Ammonia.—As previously indicated, the ammonia charge in a refrigerating system may become contaminated with impurities from various sources. These impurities are chiefly oil and water. A vacuum pumped on the system may draw water or brine through leaks in any submerged pipe. Sometimes the air used in drying the system before charging with ammonia carries moisture into the pipes and coils, and this is not thoroughly dried out. Water may also enter the machine with air through the stuffing box on the

piston rod. Although the amount of water drawn in through the box is very small, the affinity for water and ammonia is so high that whatever enters the system accumulates and at the end of the seaon's run may amount to several gallons. Grease from cutting pipe threads and oil and wax contained in packing and gaskets contribute to oil impurities. A poor quality of compressor oil is responsible for contamination by oil in a great many systems.

The ammonia charge of a plant may be cleaned by removing it into shipping drums and sending to the manufacturer for redistillation or, the charge may be purified without removal from the system by means of one of the various purifiers or regenerators.

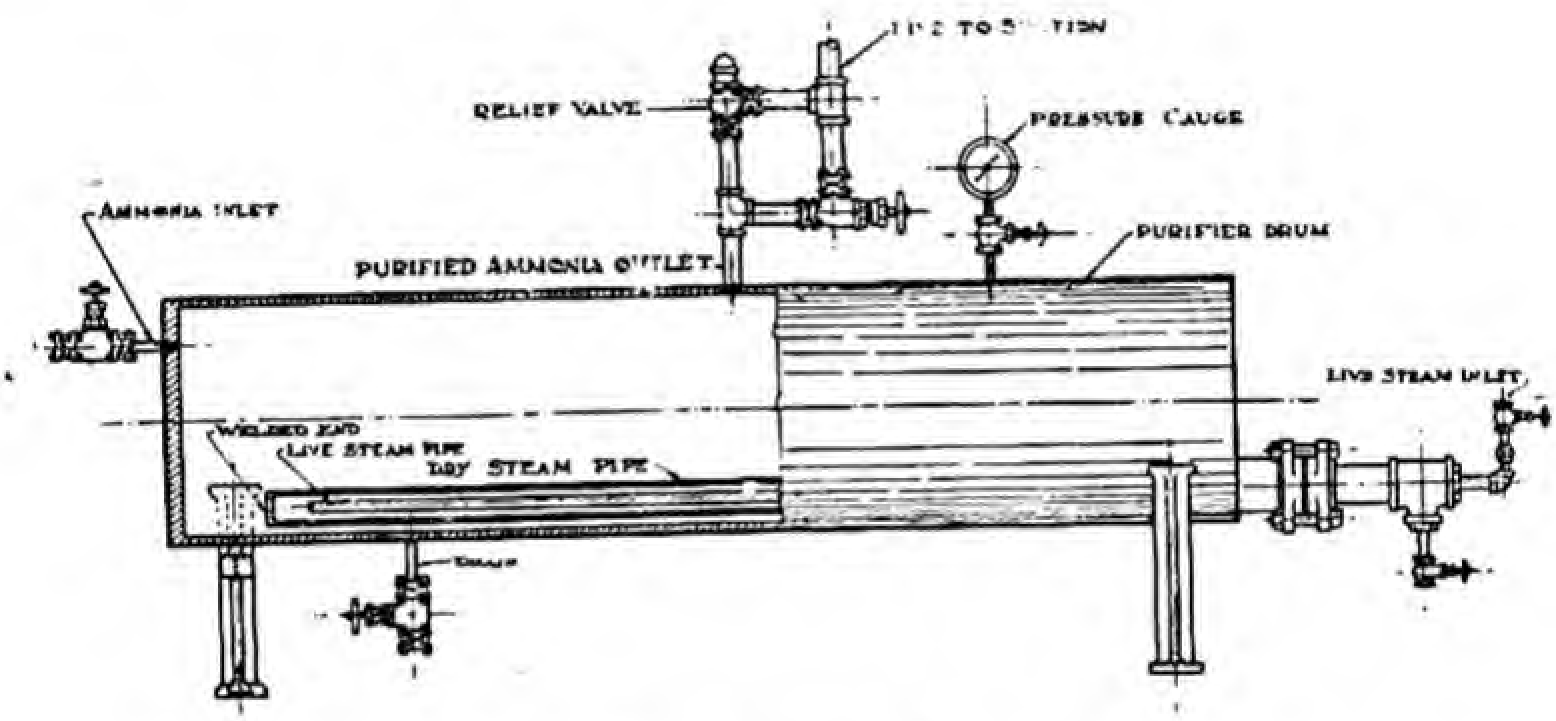

Fig. 187.—Ammonia Purifier.

Whether or not the charge is removed from the plant, each part of the apparatus should be thoroughly cleaned at least once a year. With the charge stored either in shipping cylinders or in the refrigerating coils, the condenser and receiver should be thoroughly cleaned by blowing steam through them and then, while hot, blowing air through to dry out the moisture as previously indicated. The compressor should be cleaned of an overabundance of oil and the discharge pipe from the compressor cleaned in a manner similar to the condenser coils. If the charge has been stored in the refrigerating coils, the compressor should be run very slowly, vaporizing the ammonia in the coils and thus getting it back on the high pressure side, leaving behind all the impurities possible in the refrigerating coils, which should then be cleaned out first with steam and afterwards with air. It is advisable to blow out the coils separately by disconnecting them from the headers. This will insure a better cleaning.

Some engineers may object to using steam inside of the coils and in an ice tank this may be done away with by removing the ice cans and

brine and after battening down the top of the tank and turning steam into the tank around the coils. This heats the coils up to nearly 212° F., at which point any congealed oil on the inside of the coils will be melted, and by blowing air through them the oil and dirt are easily removed.

The regenerator or purifier is designed to free the ammonia in the system from oil, water and other impurities. There are two types

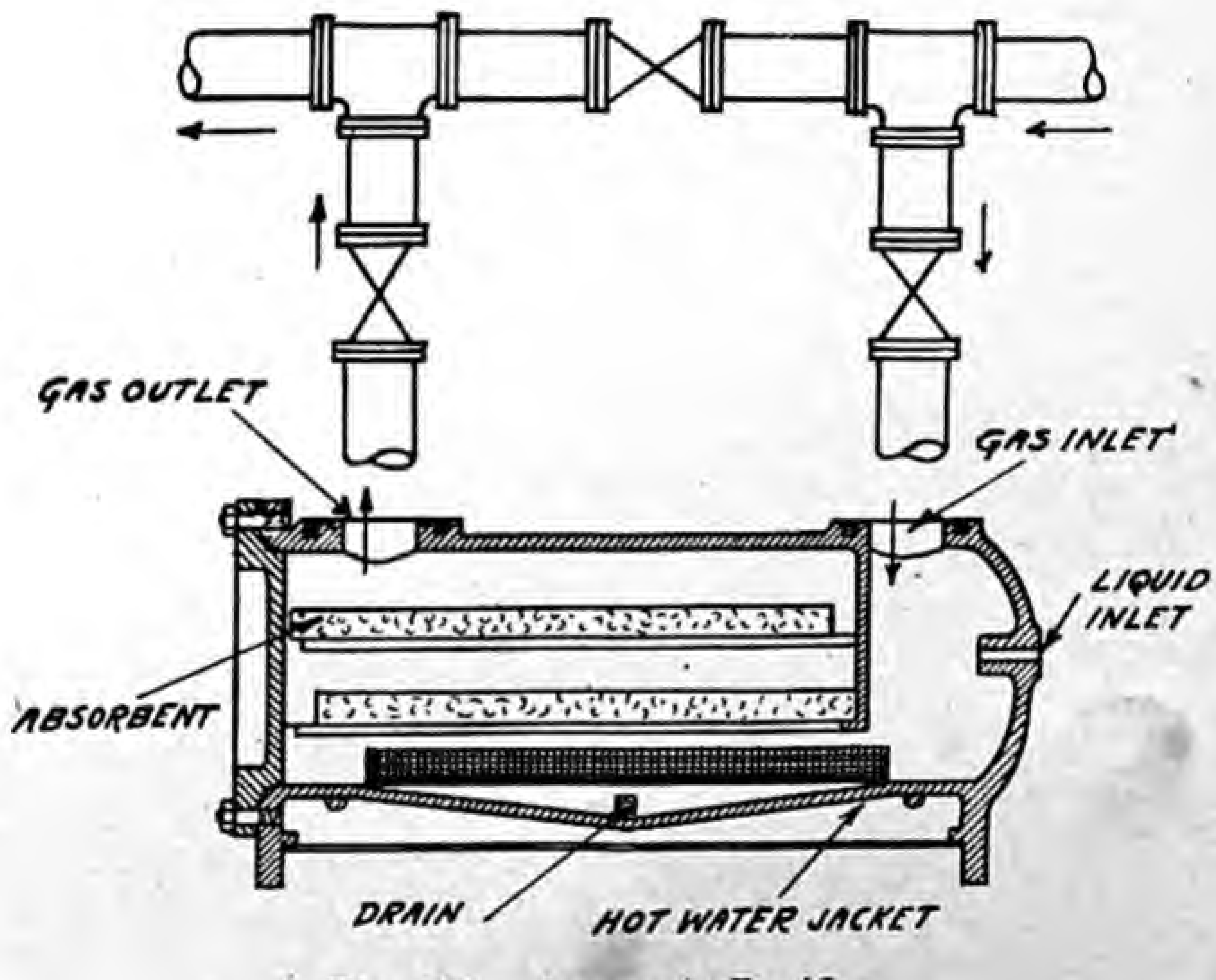

Fig. 188.—Ammonia Purifier.

of purifiers. The principle of operation of the first type is to draw a charge of impure ammonia from the receiver, oil traps or any ammonia pipe in the apparatus, and distill it. The vaporized ammonia passes to the suction line, leaving behind the impurites. In the second class of purifiers, which is strictly not a regenerator, the already vaporized ammonia, on its way back to the compressor, passes through or over trays of some drying material such as caustic soda or unslacked lime. Here it is purified or separated from its contained water and oil. Caustic soda has a very strong affinity for water and serves as well to separate other impurities, such as oil. A combination of the two types of purifiers is possible and such are made which contain a distilling or regenerating chamber in the base and the ammonia vaporizing from same passes through or over trays of caustic soda or lime. The various types of purifiers are shown by Figs. 187 and 188.

Purifying of Aqua Ammonia.—Aqua ammonia is a solvent of many of the impurities likely to exist in a refrigerating system, and to insure lively action of the ammonia, the system and the ammonia should be cleaned at least once a year.

The anhydrous ammonia in the absorption system should be drawn from the receiver into either the freezing coils or into shipping drums, and the condenser and receiver cleaned as in the compression system. If the anhydrous ammonia is stored in the freezing coils it can be evaporated after the other part of the system is cleaned and the freezing coils then cleaned in a similar manner to that described for the freezing coils of the compression system.

To clean the aqua ammonia, as much liquid anhydrous ammonia should be made as possible, thus reducing the strength of the aqua ammonia to 13° or 14° Beaumé. To remove the anhydrous ammonia into shipping cylinders, follow the method described for the compression machine, connecting the cylinder to the bottom of the receiver and exercising care that not too great an amount of ammonia is placed in each cylinder. Then the geneartor is filled with weak aqua ammonia to the operating level. All the aqua ammonia is then drawn from the absorber and exchanger into temporary storage drums. All iron rust and other impurities are now cleaned from the absorber, using a jet of steam. The lower end of the condenser, which has been previously cleaned, is now connected with the absorber and this is used to collect weak aqua ammonia distilling over from the generator when same is heated. As the weak aqua ammonia proceeds to distill gradually from the generator and is condensed in the condenser, aqua ammonia which has been taken from the absorber and is in temporary storage tanks is pumped into the generator, thus continuing distillation. When more weak and clean distilled aqua ammonia is collected in the absorber than it can hold, some of it is placed in clean shipping drums or clean tanks to be used in recharging the plant.

Corrosion Control in Brine.—If proper care is not exercised over the condition of brine it may become corrosive to various metals in the refrigerating system. Several different methods are used to keep the corrosive action of brine at a minimum. In the first place it is wise to avoid all conditions which tend to mix air with the brine. For instance, if brine is being returned to a tank after circulating through a coil the return pipe should be carried well below the surface of the brine so there will be no splashing. All other conditions where brine is splashed or air is sucked into it by whirlpools around agitators should be avoided.

In an ice tank the brine is in contact with the zinc on the galvanized

cans, as well as the steel in the coils and the tank itself. To give the best protection to these various metals the pH of the brine should be maintained in a range of 7.0 to 7.5. If all of the galvanizing is off the cans, or in a system where there is no galvanizing the pH can be maintained at 8.5 to 9.0.

In recent years it has been customary to provide additional protection of metal surfaces by treating the brine with chrome compound. Either chromic acid or sodium dichromate is used for this purpose. Both these chemicals are acid in character and will reduce the pH when added to the brine. Therefore, if the brine needs no adjustment in alkalinity, caustic soda is used along with the chromic acid or sodium dichromate to neutralize their acidity. A neutral treatment is 12 lbs. of chromic acid plus 9.6 lbs. of caustic soda or 18 lbs. of sodum dichromate plus 4.8 lbs. of caustic soda. These doses are equivalent in chromate content and either one is the amount recommended to treat 1,000 cu. ft. of brine by several authorities.

If the brine is too alkaline or contains ammonia the chromic acid or sodium dichromate can be used alone or with an adjusted amount of caustic soda to bring the pH of the brine down to the desired point. Other authorities have recommended that approximately 50 lbs. of sodium dichromate per thousand cu. ft. of calcium brine or 100 lbs. per thousand cu. ft. of sodium brine will give more complete protection against corrosion. If no pH adjustment is desired then the sodium dichromate treatment should be accompanied by 26 per cent by weight of caustic soda.

Electric Motors and Generators.*—When it is intended to run a machine for the first time, the operator should observe the following:

Make sure:

That all bolts, nuts, screws, and oil drain plugs are tight.

That the oil wells are filled with a good grade of mineral oil.

That the commutator, slip rings and other parts are clean.

That after all possible external load has been removed, the armature or rotor can be turned over by hand and runs freely.

That all external wiring connections check with the manufacturer's diagrams.

That the brushes move freely in the holders and make firm, even contact with the commutator or collector rings.

That wherever possible both motors and generators are operated initially without load (except series motors, which should be never operated with less than 50 per cent full load), for a period of several hours.

When starting motors make sure:

That the line voltage (and frequency on A. C. motors) corresponds with the voltage on the nameplate.

That the controller is in the "off" position.

That the circuit breaker (where used) or the line switch is closed first.

That the controller handle is then slowly but firmly moved on to each point of contact until it is in the running position (about 20 seconds should be consumed in this operation).

That if a compensator is used, its switch lever is moved to the starting position and when the motor comes up to speed (in about 5 to 20 seconds) the lever is quickly thrown into the running position.

When starting generators make sure:

That all resistance in the field rheostat is placed in the field circuit.

That all switches connecting the generator to any load are open.

That the prime mover is slowly started and brought up to speed.

That the resistance is gradually cut out of the field circuit until normal rated voltage is secured.

That the load is then gradually applied.

In stopping motors make sure:

First, That the circuit breaker or starting compensator is tripped or the line switch opened. Second, That the controller handle has returned to the "off" position by the time the motor has stopped.

In stopping generators make sure:

First, That the load is reduced by adjusting the field rheostat to increase the resistance in the generator field and that when the load is comparatively small, all the field resistance is inserted in series with the field windings. Second, That the circuit breaker or line switch is opened. Third, That the prime mover is shut down. Fourth, That the field switch is opened on separately excited machines.

A. C. generators which are to operate in parallel must be run at speeds which will give equal frequencies. The successful operation of generators connected to different prime movers is dependent in a great measure upon the governing of the prime movers and upon the relative fly wheel effects of the different machines.

Before two alternators are synchronized for the first time, they should be tested for phase rotation. This can be accomplished by connecting each separately to an induction motor to see if the same rotation is obtained. Another method is to throw the machines together on very low voltage with ammeters in circuit. Wide fluctuation of current on the ammeters will indicate wrong phase relation. The phase rotation can be reversed by interchanging two leads of a three-phase machine or reversing the two leads of one phase of a two-phase machine.

The usual method of synchronizing is by the use of a synchroscope. A common method, however, is to connect a lamp or lamps of proper voltage in series with the low tension windings of two transformers, the high tension windings of which are connected, one across the line and the other across the incoming machine. If the secondaries are opposed, that is, connected as for ordinary multiple connection, the main switch of the incoming machine should be closed when the lamps are dark. If the secondary or the primary leads of one of the transformers be reversed, then the main switch of the incoming machine should be thrown in when the lamps are bright. It is preferable to connect for synchronizing when the lamps are dark. The main switch should not be closed until the fluctuations in the light of the lamps become slow, that is, about one in two or three seconds.

In stopping generators in parallel the prime mover should be throttled and the field current reduced until the ammeter and wattmeter indicate a very light load. The oil circuit breaker should then be tripped, the prime mover stopped and the field switch opened.

A systematic inspection at least once a week is necessary to insure the best operation of motors and generators and the following points should be given special attention and consideration.

Both the interior and exterior of machines should be kept free from water, oil, dirt, or grease. For machines installed in very dirty places, troubles may be averted by periodically removing the rotor or armature and thoroughly cleaning the machine. A vacuum cleaner is highly recommended for cleaning assembled machines. The use of compressed air is not recommended on assembled A. C. motors. It is especially ob-

* The *Electric Power Club.*

jectionable on machines installed in foundries, machine shops, or locations where there is much carbon dust, metallic chips, etc., as the compressed air may drive the dirt or metallic chips into the windings and cause breakdowns. On direct current machines the use of compressed air is not objectionable because all the parts are more or less accessible, aspecially the commutator and brushes.

The life of bearings is affected by the lubrication, belt tension and alignment of the driving and driven shafts. Excessive wear and heating of all bearings can be reduced to a minimum by adequate lubrication, proper belt tension and accurate alignment. When bearings are unduly worn, they should be replaced, and after the new bearings have been put in, the air gap should be tested to see that it is uniform all around.

Oil wells should be filled with petroleum oil (not vegetable or animal oil) through the oil fillers while the machine is at standstill up to one-sixteenth inch of the top of the oil filler. Experience has shown that animal or vegetable oils or greases, or admixtures of them with mineral or petroleum oil will dry and gum, and by gumming ducts and oil rings, prevent the free flow of oil to the bearings. Incorrect oil level may be experienced if the oil wells are filled while the motor is running. After a motor has operated for the first week, the oil should be drawn off and the bearings washed out with kerosene, to wash out all sediment before refilling the bearings with oil. The drainage plugs should be taken out and dipped in a mixture of red lead and shellac and then replaced and tightened securely to prevent leakage. The bearings should be refilled at regular intervals, the frequency depending upon local conditions, such as cleanliness, severity or continuity of service, etc. After changing the oil, the oil rings should be always inspected to make sure that they are in the their proper position and turn freely.

The brushes should be inspected to see that they move freely in the holders and at the same time make firm, even contact with the commutator. The brush tension should be checked regularly to make sure that the proper tension (1¾ lbs. to 3 lbs. per sq. in.) is maintained by the spring.

When replacing brushes, they should be fitted by means of fine sand paper folded around the commutator and the rotor revolved by hand in the desired direction until a proper fit is obtained. On some machines, the sand paper can be held in place if it is cut to a width slightly narrower than the commutator, and the front end of the strip inserted into one of the narrow slots between commutator bars (where the mica has been undercut), and then folded back around the commutator by slowly revolving the armature by hand until the paper moves under a set of brushes.

An operator should never shift the position of the brushes unless he knows positively that the brush position is incorrect or it is desired to change the direction of rotation or mounting of the machine. On machines with commutating poles, the position of the brushes is fixed on the neutral point at the factory and the position of the brushes on such machines should never be shifted except to make change as indicated in the previous sentence or for compounding or parallel operation of generators.

Care should be taken to see that the pigtails or flexible copper conductors are firmly fastened in place so that they will carry their full current from the brush to the brush holder. A slight extra length should be left in the pigtails because if they are too tight they might tend to pull the brush out of line and out of proper contact with the surface of the commutator.

Commutators should be always kept clean and well polished. Under normal operating conditions, a commutator will require only occasional cleaning with a piece of canvas or non-linting material. No vaseline or oil of any description should be used on a commutator. In case the commutator has become rough, this roughness may be removed by polishing the commutator with a piece of sandstone, from which a segmental piece has been cut, having the same radius as the commutator. If sandstone is not easily obtained, sand paper may be used by pressing it against the surface of the commutator with a block of wood shaped like the standstone mentioned above. In both cases, the commutator should be run at a high rate of speed during polishing and the sandstone or sand paper moved back and forth along the surface parallel to the shaft. After this has been done, the commutator and brush faces should be carefully cleaned to remove any grit which might cut or scratch the commutator. Emery cloth should never be used on a commutator or brush.

It is advisable to apply a little clean lubricating oil to the collector rings regularly. The oil may be applied with a piece of cloth of chamois skin.

When a machine appears to be too hot to the touch, the actual temperature should be measured with a thermometer. The safe operating temperature of a motor or generator cannot be accurately determined by the hand, as it is impossible to hold the hand comfortably on a motor which has been heated to within its guarantee. The temperature rise for which a machine is guaranteed at its rated load is shown on its nameplate. The guarantee is based on the surrounding air temperature, assumed at 40° C. (104° F.) or less and the machine can be operated safely if its temperature by thermometer does not exceed 90° C. or 194° F.

QUESTIONS ON CHAPTER XVIII

1. Define natural and functional depreciation.

2. What is the proper method of procedure in repairing and overhauling a refrigerating plant?

3. What are the principal parts about a compressor which require inspection and repairs, and how are these parts kept in adjustment?

4. What is one of the most important factors in the operation of a condenser?

5. Explain how you would overhaul and repair an atmospheric type condenser.

6. Explain how you would overhaul and repair a double pipe type condenser.

7. How frequently should a condenser be cleaned of sludge or scale?

8. How are the different types of ammonia evaporators kept clean, and why is it so important to keep these surfaces clean?

9. How are anhydrous and aqua ammonia purified in an absorption system?

10. Describe the various methods of corrosion retardation.

CHAPTER XIX.

OPERATION AND CARE OF APPARATUS* (Continued)

Packing.—When a moving part of a machine or apparatus passes through a wall of the machine or apparatus into a region of higher or lower pressure, a packing is interposed between the stationary and the moving parts in order to reduce to a minimum the leakage of the medium from the higher pressure to the lower pressure. The importance of the proper selection and care of packing may be more thoroughly appreciated when one remembers that approximately 75 per cent of the loss of ammonia in the average compression refrigeration plant may be charged to leakage through the stuffing boxes. It is probably true that there is no other part of the system that requires so much regulation and care, and causes so much anxiety as the stuffing-box. The difficulties that are generally encountered are largely due to excessive friction of wide ranges of temperature.

Principle of Packing.—When two solid surfaces are held together by any appreciable force, any effort tending to move them relatively to each other—that is, to move or rub one surface over the other—is met by a resisting force acting tangentially to the surface of the two bodies. This resisting of relative motion, commonly termed friction, is due to the interlocking action of the minute depressions and elevations which exist even in the smoothest surfaces, and will vary with the different kinds of materials, finish of surfaces and the speed of the relative motion. The appearance of these irregularities of the surfaces is illustrated by Fig. 189. This figure shows a small section of the separating surfaces between a smooth piston-rod and a smooth packing substance, as it would appear under a magnifying glass. It is apparent that if these two surfaces are held in contact by a force, the effort required to slide one over the other is quite large.

Separating Oil Films.—If oils or lubricants are interposed between the surfaces, the frictional resistance is materially reduced because

* Considerable matter in this Chapter has appeared in *Ice and Refrigeration* and *Power* at various times, and permission has been granted for its re-use here.

the rubbing surfaces are wholly or partly separated by a thin film of the lubricant. Fig. 189 shows how the film of oil separates the packing and rod so that they do not tend to seize. It is obvious that if the rubbing surfaces are made as smooth as possible the frictional resistance to motion will be lessened and the surfaces will be easier to lubricate. If the depressions are filled with some suitable substance and the surface glazed over smooth, the cause of frictional resistance will be partly overcome. Fig. 189 shows how a packing and rod may be glazed over with graphite and separated entirely by means of an oil film.

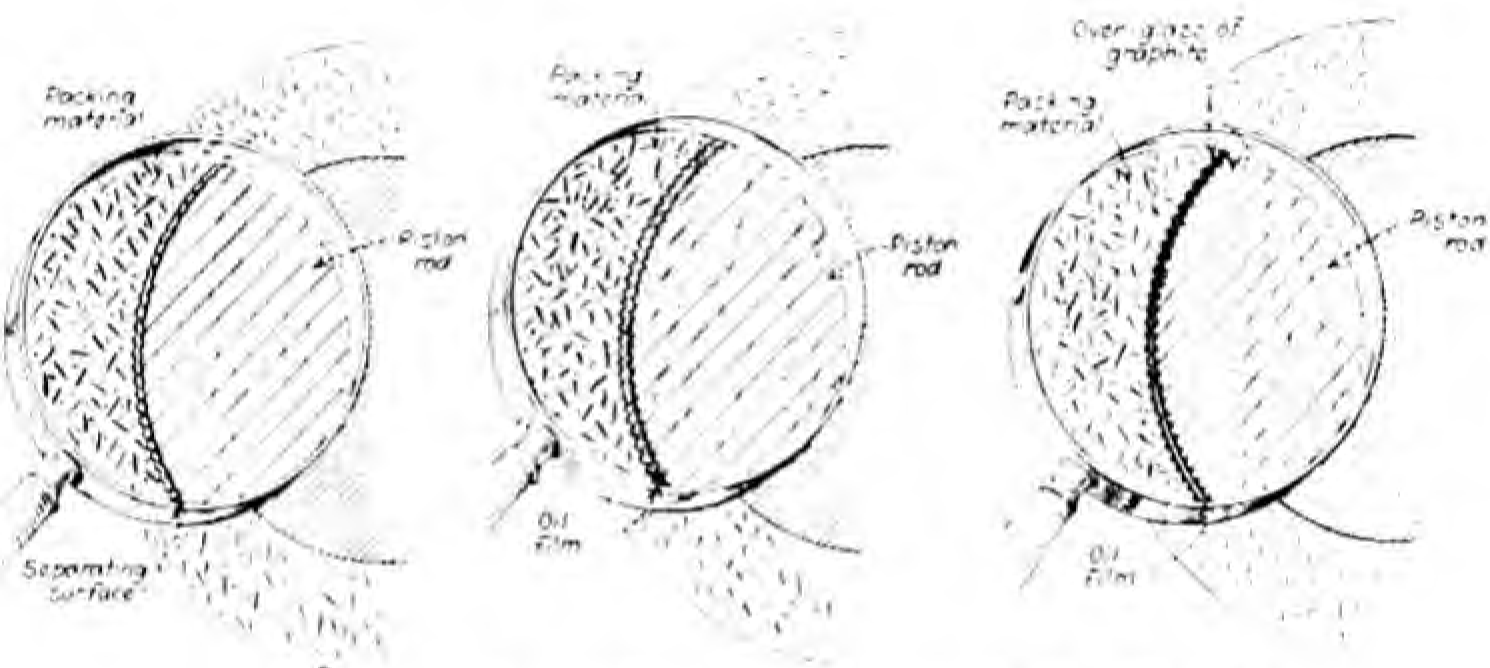

Fig. 189.—Oil and Graphite Films on Piston-Rod.

The proper functioning of the stuffing-box depends upon the presence of this oil film. Owing to the phenomena of capillary attraction and molecular adhesion, the oil film clings to the surface with such tenacity as to support a load as large as 800 lbs. per sq. in. The exact magnitude of this load intensity is governed by a number of things, such as character and type of surface, finish of surface, kind of oils, etc. It is apparent that if a film of oil is interposed between the packing and the piston-rod, as illustrated by Fig. 190, and that if the packing is subjected to pressure due to forcing the gland into the box, the escape of high-pressure gas from A along the rod to the atmosphere at B is opposed by the resistance of the oil film. This is the fundamental principle of all successful packings. Since the rod is in motion and under vibration, is is necessary to have a considerable length of effective film between C and D in order to prevent the escape of gas of different pressures.

It should be further noted that the employment of frictional substances as packing is inconsistent with sound engineering principles, since a packing consisting solely of frictional materials depends upon the pressure of the substance against the face of the rod to prevent the escape of the medium under pressure.

The effect of varying temperatures of the rod upon the packing should be observed. The compressors in actual operation are subjected to different temperatures due to operating on wet, dry, or superheated vapors. The packing should respond to slight changes of dimensions instantly in order to conform to the rod. This would preserve the oil film and prevent the escape of gas. The packing should then be made of some resilient material that will compensate automatically and instantly for any irregular movement of the rod, or any expansion or contraction due to changes of temperature.

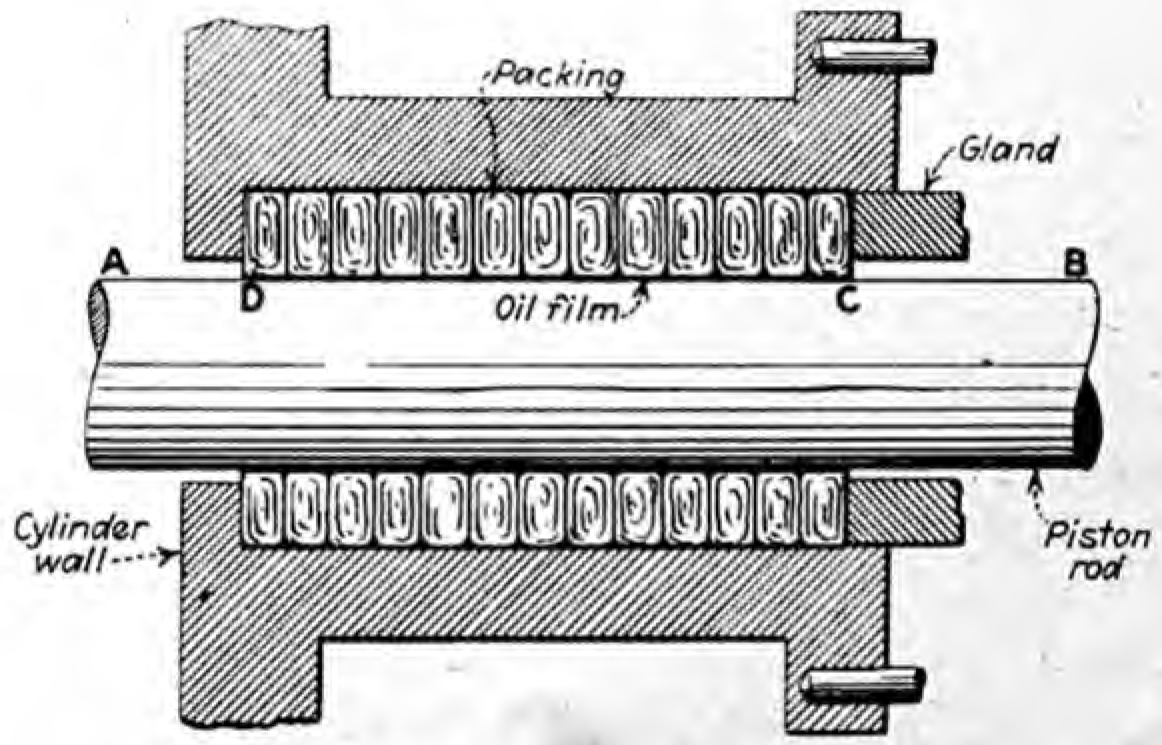

Fig. 190.—Compressor Stuffing-Box.

Requirements of a Good Packing.—From the foregoing it will be appreciated that to be successful a packing has certain definite requirements that must be fulfilled. It will be observed also that the desired functions of the packing are determined by economic considerations. The following may be considered cardinal properties of good packing:

In the first place, the packing must be gas-tight. It must reduce to a very small minimum or prevent entirely the escape of the medium under pressure. The packing must remain gas-tight under the variations of temperature that occur in actual operation. The escape of the refrigerant increases the cost of operation, since new ammonia must be charged into the system to make up for losses.

In the second place, the frictional resistance of the piston-rod, as it reciprocates to and fro through the packing-box, should be reduced to a minimum. It is apparent that the energy used to overcome frictional resistance is simply wasted. This energy cannot produce any good and cannot be recovered; therefore, it is desirable to keep this loss as low as possible. Furthermore, excessive friction causes great wear and abrasive action on the surface of the rods, so that they become

scored and shouldered. Under this condition larger quantities of lubricating oil are required.

In the third place, the packing must be resilient. It must be sensitive to any change of pressure and temperature or any irregular movement of the rod. It must adjust itself automatically and instantly to the operating condition that exists.

Lastly, the packing should have a fairly long life in order to reduce renewals and save money. A packing that will function properly for the longest possible time should be selected for the service.

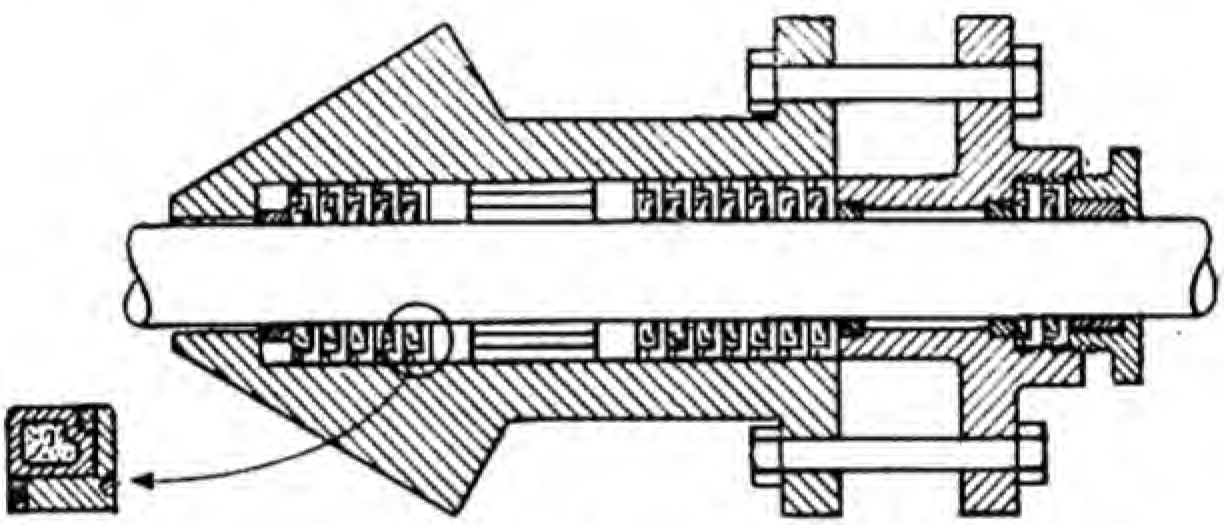

Fig. 191.—Non-Elastic Packing in Stuffing-Box.

Types of Packings.—There are many types of ammonia packings that must be used to pack the stuffing-boxes of the compressors, pumps, and valves in refrigerating plants. In order to make an intelligent selection and adapted to a particular case one should be familiar with the construction and merits of the various types that may be purchased in the market at present. Among the many types which are available at present the following are most important: Collapsible tube metallic, flexible metallic, elastic metallic, fibrous or soft, and combination types.

Collapsible Tube Metallic Packing.—Fig. 191 illustrates the application of non-elastic metallic packing in a stuffing-box. This type may also be termed a collapsible tube type. The application of pressure by the gland slightly collapses the tubes in the direction of the length of the rod, which compensates for wear and holds the face of the tube against the rod. The packing is made up of a series of hollow babbitt rings filled with graphite. The babbitt rings are supported by cast-iron cages which clear the rod. Both the ring and the cage are split and the joints are sealed by the use of gaskets. These gaskets also provide for a slight amount of resiliency. The graphite flows from the babbitt rings through the emission holes due to the action of

the rod, keeping the rod and packing glazed over with a thin layer of graphite at all times. A series of these cast-iron and babbitt rings are placed in each compartment of the stuffing-box.

The collapsible tube-type packing meets the requirements of a good packing in an efficient manner. In actual operation it reduces the leakage to a minimum. Since only babbitt metal touches the rod, the work required to reciprocate the rod to and fro in the box is low. It is self-adjusting within reasonable limits of change of conditions. It is quite durable in service, the useful life of the packing being from two to ten years, with an average life of five to six years. The useful life of the packing depends upon a number of variables, of course.

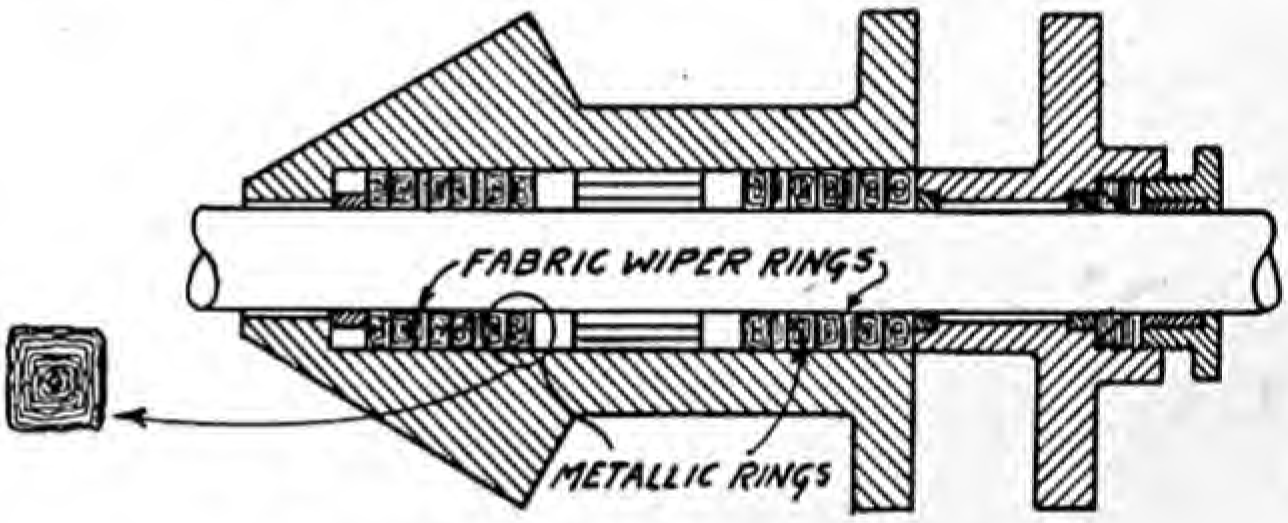

Fig. 192.—Flexible Metallic Packing.

Being of rigid construction, the packing required delicate adjustment of the gland to secure proper pressure upon the rings. Also, as the lower crosshead slide and the bottom of the cylinder become worn down, the center line of the piston rod is lowered, which results in concentrating the weight of the rod, ètc., upon the packing. Under this condition the metallic rings tend to wear to an elliptical form, which makes leakage probable. This type of packing may be used with good results on all types of machines, horizontal double-acting, vertical single-acting, enclosed, etc. When used on aqua ammonia pumps, the graphite prevents the rods from pitting and corroding.

Flexible Metallic Packing.—Another form of non-elastic metallic packing is illustrated by Fig. 192. This type is commonly termed "flexible metallic packing." It is made up into sprials of continuous convolutions of babbitt ribbon passed through a bath of lubricant. Rings of this metallic packing are cut off and assembled with fabric wiper rings on the rods as indicated.

Another form of this same general type of packing is largely used in refrigerating plants. In this type, V-shaped solid babbitt rings are backed up with the flexible metallic rings and fabric wiper rings are also inserted between the metallic rings.

The flexible metallic packing is an efficient packing. The packing reduces leakage to a minimum due to the fact that babbitt rings are held firmly against the rod by the pressure exerted by the gland. The packing is compressible and, therefore, permits of the laternal movement. Amalgamation under pressure is prevented since there is a thoroughly broken metallic surface between each convolution of the babbitt ribbons. The friction loss is small, since a large portion of the packing against the rod is composed of anti-friction metal. Therefore the rods run cool. The packing adjusts itself automatically in compensating for expansion and contraction due to temperature changes. No gland adjustment is necessary excepting a slight take-up to compensate for wear. Additional rings must be inserted at certain periods to make up for the wear of the rod on the packing. The life of the packing, however, is affected by the physical condition of the rod. It gives the best service on rods that are true and in line. On account of being supplied in the form of handy spirals, and being flexible, it is easily installed. This type is used for service on the rods of any compressor having a speed of 400 ft. per min. or less.

A great portion of all packing in any plant is used on the hand valve stems. Flexible metallic packing has often been used for this service with marked success. Rings are cut from the spirals and are placed in the boxes around the stems, breaking the joints. This packing prevents leakage, permits the stems to rotate freely, and protects them from scoring, rusting or pitting.

The flexible metallic packing, combined with solid V-shaped babbitt rings, may be used on any compressor rod having a speed from 400 to 1,000 ft. per min. The solid babbitt rings prolong the life of the packing. Otherwise it has the same general advantages as the regular flexible type.

The flexible metallic packing with a fabric backing has been produced to supply the demand for a flexible metallic packing to be used on rods that are slightly out of true and out of line. The resilient backing compensates for the irregular physical condition of the rods. This type should not be used on rotating shafts on account of the fact that scoring would be produced by the action of the fabric backing upon the rod. It produces a gas-tight joint on the surface of the rod. Due to the insertion of the fabric backing the friction loss is increased slightly. On the other hand it is automatic in adjustment to change of conditions. It is made of durable materials.

Combination Packing. — A combination of metallic and fibrous types of packing is used sometimes. Babbitt rings are held against the surface of the rod by the fabric rings when pressure is applied by means of the gland. The primary rings are supported by the metal

backing rings. Suitable gaskets are inserted for sealing the joints in the rings.

The combination packings have been produced in order to supply a packing that would combine the advantages of both the fibrous and metallic packings into a single packing without incorporating the disadvantages of either type. They depend upon the action of the metallic oil seal to prevent leakage rather than the pressure of the packing against the rod. Friction losses are low since the rubbing surfaces are composed of anti-friction metals. They are automatic in adjustment to physical conditions on account of the resiliency of the fabric portion of the rings. On account of being comprised of durable materials and on account of having solid metal rings against the rods the wear on the packings is slow. The packings may be used on all types of compressors. They may be used on rods that are not true and out of line slightly.

Elastic Metallic Packings.—The elastic metallic type of packing has been used in steam engines to a considerable extent and is being used at present in ammonia compressors. The metallic rings are held against the surface of the rod by helical springs around their circumferences. The rings are made into three segments and make tight joints on the rod and against the partitions of the casings. The rings and casing are usually made of cast-iron. The rings and casings are machined and fitted to the individual boxes and rods. Fibrous wiper rings are inserted in the gland to retain the lubricating oil.

The metallic packing of the elastic type has been used extensively in certain fields with apparent success. The oil film between the cast-iron rings and the rod prevents the escape of the high-pressure gas. The friction of the special cast-iron rings on the rod is quite low. In fact, it is claimed that this type of packing requires but one-quarter of the amount of power to overcome friction that the fibrous or soft packings demand. The packing gives the better service on the larger rods. The packing has no means of adjustment after installation. Each set is machined and fitted for the particular rod on which it is used. Therefore it is limited to exclusive use on the original rod. However, it operates efficiently under changing conditions. It has a very long life, lasting from five to twenty-five years. Its life is affected by the wear at the various joints.

This type of packing has been used extensively in the past on high duty apparatus. It may be used on rods of all types of compressors. The rods should be true and in line for the most efficient operation, but it will function properly on rods that are slightly out of line, since the rings may "float" in the compartment of the casing. On the other hand it is not suitable for all vapor and liquid conditions that may

occur in the refrigeration plant. It gives best service on compressors which handle dry vapors that are only slightly superheated. This condition may be obtained oftentimes by the proper adjustment of the liquid expansion valves. It should be well supplied with lubricating oil at all times. The first cost is comparatively high on account of the accurate machine work that is required in the fabrication of the sets. It has been used on the largest steam, air, gas and ammonia plants, under all conditions of pressure and temperature.

Fibrous or Soft Packing.—Fibrous packings are generally composed of a combination of rubber and a fabric or cloth made of cotton, flax or asbestos. These are vulcanized and thoroughly lubricated and are made up into several forms, such as round, rectangular, or wedge-shaped, with rubber cores and backs and with laminations being horizontal, vertical, or diagonal. The function desired is that the packing will move toward the face of the rod when pressure is applied by the gland.

Fibrous packing is composed of a special grade of rubber and duck and is thoroughly lubricated. The laminations are vertical. This type may be cut up into sectional rings. The application of this type of packing to a compressor stuffing-box is indicated in Fig. 193. The sectional rings are interposed between the plain rings at the ends of each packing compartment.

Another form of packing may be made by using the regular fibrous packing rings and fibrous section rings and by alternating the two types as they are placed into the box. This method of packing the box is shown by Fig. 194.

The relative position of the gas lantern in both cases should be noted. Fig. 193 shows the desirable arrangement, that is, a greater number of rings should be placed between the gas lantern and the region of high-pressure.

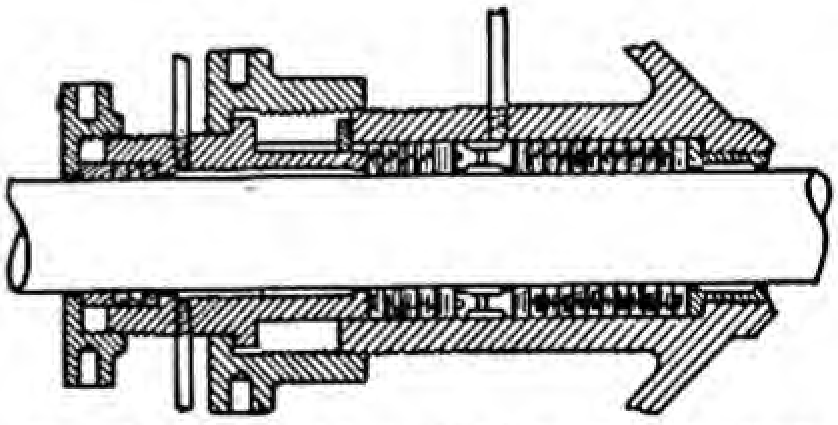

Fig. 193.—Sectional Fibrous Packing.

The soft or fibrous packings perform their functions quite well providing they are under the care of a careful operating engineer. They prevent the leakage of other medium under pressure by being forced against the surface of the rod by the gland, since they have lateral movement due to being comprised of resilient materials. On account of being composed of materials that are more or less frictional in character and because of the forcing of the packing against the rod, the power required to overcome the frictional effect becomes excessive, if the packing is not given the proper attention. The soft rings contain

lubricants but under operating conditions a liberal use of oil is necessary. Sometimes thin lead washers that clear the rod are inserted between the fibrous rings to facilitate lubrication. The excessive friction effect necessarily causes the scoring, wearing and shouldering of the rods. Bits of the packing may find their way into the cylinder. Ordinarily this type of packing is not sensitive to the change of temperature. Thus, when the machine becomes cool, it may be necessary for the operator to tighten up the packing gland to prevent leakage, or if the machine is allowed to become hot, it is necessary to loosen the gland in order to prevent harm to the packing due to the excessive heating effect of the frictional work. In consideration of these facts it will be noted that the life of the packing depends upon the care given to it. Furthermore, it will be noted that on account of the friction between the packing and rod surface, and since the materials comprising the packing may disintegrate under operating conditions, the useful life of the packing is short. The fibrous type packing has been used extensively on all types of compressors and pumps. It has been used on rotating shafts, but a decided grooving action may develop if the packing is not properly regulated.

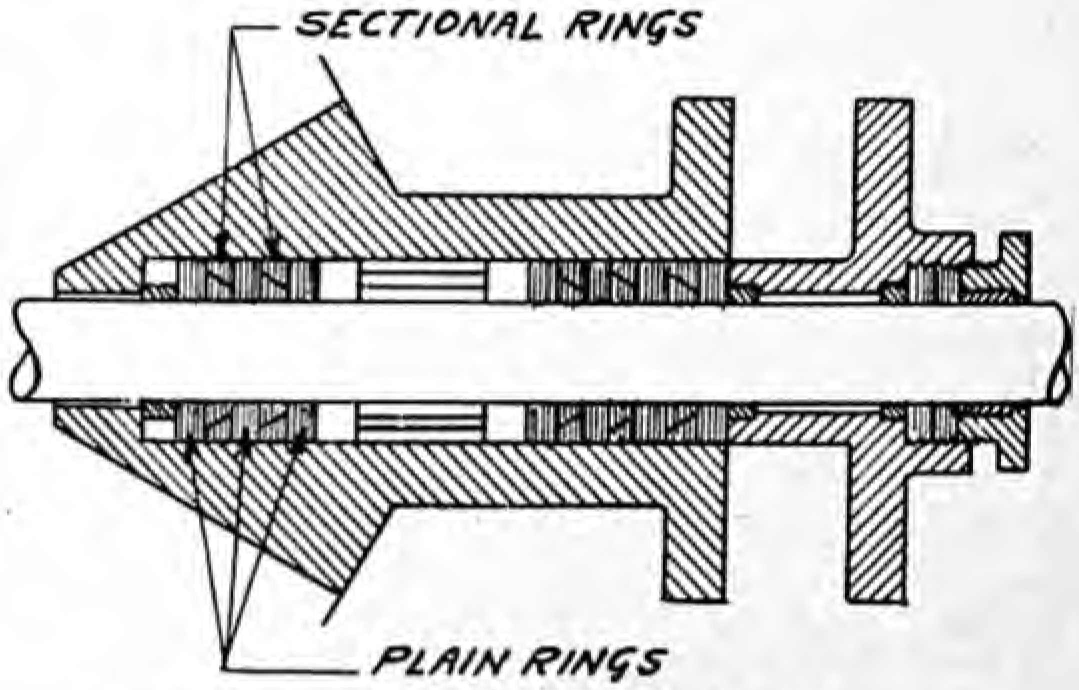

Fig. 194.—Sectional and Solid Fibrous

In the selection and application of packing to refrigerating machines it is advocated that each packing problem be solved intelligently from all viewpoints. It is reasonable to expect that due consideration must be given to such factors as the type of machine, conditions of operation, the kind of operating engineer, physical condition of the rod, the cost of the packng, the worth of the possible saving of ammonia, and so on. Procedure along these lines of action will do much to the elimination of packing troubles.

The Merits of Packing.—The metallic type packings have been used generally in the past for service on high-duty apparatus. This type has been found to be dependable and therefore successful in this field. No doubt the local conditions have had an important bearing on the efficient operation of this packing. Since this type depends on the establishment and maintenance of an oil film between the packing

substance and the rod, it must be well-supplied with lubricating oil at all times. Also it is necessary that the rod be true and well aligned. Therefore this type has been in a more or less advantageous position to effect efficient service.

The fibrous, or soft packings, have been applied usually on the low-duty apparatus, such as pumps, valve stems, plungers, shafts and piston-rods. The field of application of fibrous packing is relatively large, since there is much more low-duty equipment than high-duty apparatus. This type of packing gives desirable results on rods that are not true. A large amount of efficient service has been obtained from this kind of packing in the past and the same performance may be expected in the future.

The metallic type may be considered somewhat more gas-tight than the fibrous type. The gas-tightness of the former depends on the oil seal, while the latter depends more or less on the physical contact of the rod and packing material. The metallic type has less friction on the rod, since the greater portion of the packing that is against the rod is constructed of anti-friction or bearing metals. Also this type depends on the oil seal to prevent the escape of the medium under pressure rather than the forcing of the packing against the rod to stop the leakage of the gas, which results in reduced friction. The soft packing is comprised of such material as rubber, cotton fabric and asbestos, which have large coefficients of friction. In any case too much packing should not be placed in the box, so that when the gland is started squarely, excessive friction is not produced by the extreme pressure created by the gland. The gland nuts should be only hand-tight.

With changes of temperature within the cylinder the metallic type is more or less automatic or self-adjusting in compensating for the expansion and contraction of the materials. On the other hand, the fibrous packing will expand and contract with changes of temperature so that leakage will occur if the glands are not readjusted. However, the metallic types (exclusive of the elastic type) require a much more delicate and careful adjustment of the gland than the fibrous types do, in order to secure the proper pressure on the packing.

Generally speaking, it may be said that the metallic packings are more durable than the fibrous kind. This is due to the fact that the fibrous packings are composed of materials that tend to disintegrate under the actions of friction and heat. In the ultimate analysis it will be noted that the operating engineer is responsible to a large extent for the successful or poor performance of the packing materials, since it is his duty to give to this important part of the compressor its proper care, adjustment and lubrication.

Lubrication.—The primary function of the use of oils and other lubricants about the refrigerating plant is to provide for lubrication of moving parts. The oil or other lubricants accomplish this purpose by forming a thin separating film between the two moving parts. This film of oil or lubricant is very thin and serves to keep the various parts of the moving surface from touching each other. The establishment of an oil seal or film between the working parts reduces the friction very materially. The surfaces of the moving parts are really made up of minute depressions and elevations, and it is evident that when these surfaces come into direct contact, the friction or the force required to move these parts relative to each other is much larger. In order for the fluid to function as a lubricant it should have the minimum cohesion between the particles of itself and maximum adhesion to the surfaces to which it is applied. It should remain fluid at the usual working temperature of the bearing surfaces. It is evident that if the bearing gets quite warm and if the lubricant has a low evaporating point there will be considerable loss of lubricant due to the evaporation of same in the bearings, stuffing-boxes, etc. The particular kind of lubricant to be used for any particular purpose will depend upon several variables such as the character of the friction, the various pressures and temperatures, the velocity and disposition of the moving parts, the bearing pressure exerted between the moving parts.

In addition, attention must be given to the chemical properties of the oil, giving particular attention to the fact of whether or not the oil will attack the metals used in the bearings or in the oiling system. Also consideration of the viscosity of the oil is important. Generally the desirable viscosity of the oil is determined by the mechanical condition of the parts for which it is used for lubrication. Thus, if it is desired to lubricate an ammonia compressor cylinder, the mechanical condition of the compressor valves, the piston rings, etc., will determine, in a large measure, the viscosity of the oil to be used.

Oil for Refrigerating Plants.—The selection of a suitable oil for the various parts of ice making and refrigerating plants presents more difficulties than are encountered when selecting oil for a steam power plant. The oil must have a comparatively high flash point, ranging from 300° to 400° F. On the other hand, it is evident that the oil should have a low congealing point. This is due to the fact that it is impossible to keep some of the oil from getting into the evaporating coils, under which condition, if the oil has a comparatively high congealing point, it will accumulate on the cold surfaces, thereby reducing the heat transmission rate of the surface materially. The exact congealing point will depend upon the relative temperature used in the evaporating coils. In general the congealing point of oil used in the

refrigerating plant varies from $+5°$ to $-40°$ F. Straight mineral oils are used largely for lubricating purposes in the refrigerating plant. These oils are highly refined and filtered straight run distillates. The mineral oil does not show a tendency to absorb moisture so readily and it has a comparatively lower setting point than the compounded oils. The compounded oils are those which have been compounded with an animal or vegetable oil.

In addition, the straight mineral oils do not seem to saponify as readily as the compounded oil. The acidity should receive attention also. Generally not more than 0.03 milligrams of potassium hydroxide should be required to neutralize one gram of the oil. The oil should separate completely in thirty minutes from emulsions made with distilled water, one per cent salt solution or normal caustic soda solution.

Properties of Oils.—In oils for refrigerating machines the principal properties to be considered are: the flash point, the viscosity range, and the cloud and pour points.

The flash point is the temperature at which the vapor above the surface of the oil will flash or momentarily ignite when a flame is applied to the vapor. The fire test is the temperature at which the oil will burn continuously when ignited. The fire and flash points are usually determined by an open cup flash and fire tester in the laboratory.

The determination of the viscosity is another laboratory test. The viscosity is the measure of the fluidity of the oil. The viscosity is usually determined by the Saybolt Standard Universal Viscosimeter. The viscosity as determined by the Saybolt Viscosimeter is the time required in seconds for 60 c.c. of oil to flow through a given standard orifice. As the viscosity of the oil will vary with the temperature, it is customary to compare the oil at a given temperature. Oils for the refrigerating plant will generally be tested at a temperature of 100° F., while heavy oils for steam cylinders, motors, etc., will be tested at a temperature of 210° or 212° F.

The relative viscosity of the oil may be determined in a practical manner as follows: A small glass tube with a contracted end, such as a medicine dropper, is filled to a certain depth with oil. The number of drops falling from the tube in one minute is an indication of the relative viscosity of the oil. Or, drops of oil may be placed on the edge of a piece of glass, after which the glass is tilted. The time required for each drop to reach the lower edge of the glass will give a means of comparing the relative viscosity.

Of especial importance in the ice making and refrigerating plants are the cloud and pour test of lubricating oils. The pour test of an oil is the lowest temperature at which the oil will flow. The temperature

of the pour test is usually taken as being 5° above the setting or congealing temperature. Paraffine oil will have a higher pour test than the asphaltic base oils, due to the presence of a small percentage of paraffine wax in the oil. Paraffine wax has a melting temperature of 110° to 115° F., and when in solution with a lubricating oil it will cause the oil to congeal at a comparatively high temperature. The temperature at which the oil becomes cloudy due to the separation of the paraffine wax is called the cloud test.

Fig. 195 shows a practical method that may be used to determine the cloud and pour test by the operating engineer.

The freezing mixtures commonly used are as follows:

For temperatures down to 35° F., ice and water.
For temperatures down to — 5° F., crushed ice and sodium chloride.
For temperatures down to —25° F., crushed ice and calcium chloride.
For temperatures down to —70° F., solid carbon dioxide and acetone.

If refrigerating compressor oil does not have a sufficiently low congealing temperature it tends to remain in the evaporator and is difficult to remove. Therefore the pour test is important.

It is sometimes desirable to make an evaporation test of oil. To do this, a small quantity is placed in a shallow dish and carefully weighed. The dish and the oil are then exposed to a temperature of 212° F. for four hours, after which the dish and oil are weighed again. The loss should not be more than 0.25 to 0.50 per cent. The most important thing derived from the evaporation test of oil is the condition of the oil after it has been exposed to a higher temperature. The one having a greasy, oily residue will be the one most suitable for lubrication, and the one having a sticky or tough res-

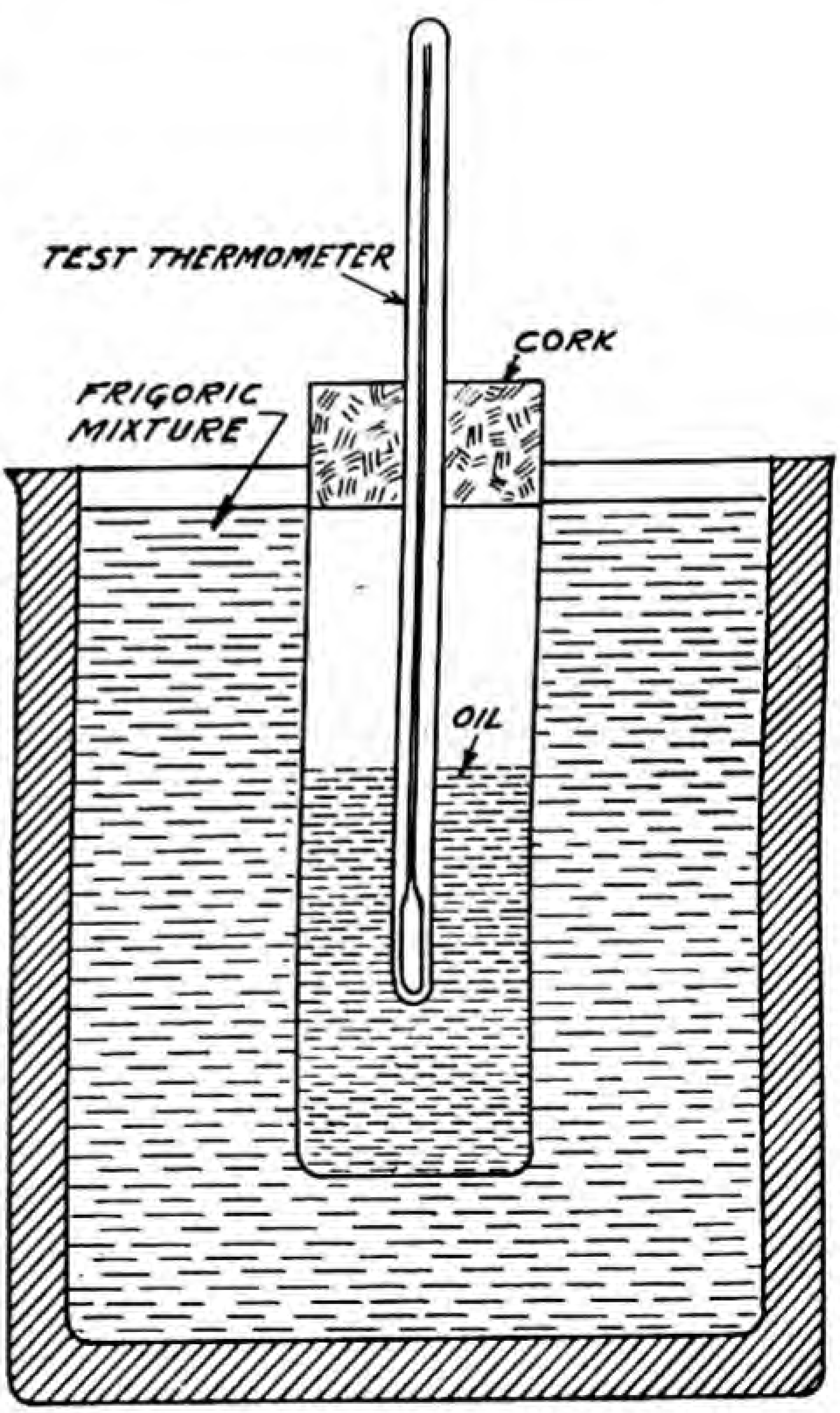

Fig. 195.—Practical Apparatus for Testing Oil.

idue should be rejected, because such oil would produce deposits in the cylinder, thereby giving unsatisfactory results.

The acidity of the oil should be carefully noted. It must be remembered that the function of the oil is to produce a separating film between the highly polished surface of the bearings. From this it is evident that the oil must not attack the metals of the bearings. Sulphuric acid is used during the preparation of the oil and may be present in small quantities. The presence of sulphuric acid may be determined by litmus or phenolphthalein paper after the oil has been thoroughly washed with warm water. Another simple test for acid in the oil is to immerse in a sample of the oil a piece of highly polished copper for 24 hours. There should be no change in the brightness of the polished surface at the end of 24 hours. If acid is present, the surface will be dulled.

Oil for Compressors.—Ammonia compressor cylinders of the open type, either in horizontal or vertical position, may be lubricated with oils having Saybolt viscosities ranging from 130 to 180 at 100° F.; setting points varying from +5° to =25° F.; with flash points varying from 325° to 400° F. The bearings of this type of machine may be lubricated with oil having a viscosity varying from 175 to 250 at 100° F. Carbon dioxide cylinders of the same construction may be lubricated with an oil having a Saybolt viscosity of 130 to 180 at 100° F., a setting point of −25° F., and a flash point of 325° to 360° F. The bearings of the open type carbon dioxide machine may be lubricated with oil having a Saybolt viscosity varying from 175 to 250 at 100° F.

The small vertical enclosed type compressor using either ammonia or carbon dioxide should be lubricated with an oil having a Saybolt viscosity ranging from 60 to 70 at 100° F., a setting point of −40° to −50° F., and a flash point of 300° to 325° F. The larger vertical enclosed type compressor, using either ammonia or carbon dioxide, may be lubricated with an oil having a Saybolt viscosity of 130 to 180, a setting point of −25°, and a flash point of 325° to 360° F.

Ammonia compressor cylinders of the open type construction are generally lubricated through the stuffing-box. The small vertical single-acting enclosed types of compressors are sometimes lubricated by the splash oiling system. The larger type of vertical enclosed type compressors are lubricated by the positive force-feed oiling system. The forced-feed oiling sytem is sometimes installed on the larger compressors of the open type of construction. The pressure required for forcing the oil to the various rubbing surfaces is generally supplied by means of a pump or an overhead oil tank. The system is more or less automatic and consists of an oil shelter, storage tank, water separator, oil pump, oil connections, and sight-feed oil valves.

In the open type of machine, all of the moving parts are protected by steel guards which are removable, the purpose of this being to prevent as much as possible the waste of oil. Instead of using the central oiling system as previously described, the compressors of the open type may be equipped with manually or mechanically operated lubricators. Thus, the stuffing-box may be supplied with a pump which may be operated either by hand or by connection to some moving part of the compressor. Adjustable sight-feed oilers may be provided for the crosshead guides and the main bearings of the machine. The wiper, oiler and cup, or one of telescopic design, may be provided for the crosshead pin, while a centrifugal oiler of the pendulum type may be installed to supply the oil for the crankpins.

Anhydrous ammonia does not seem to affect the oil that is allowed to get into the compression system at such temperatures which exist in this type of system. However, in the presence of such impurities as water, etc., ammonium hydroxide may be formed which will have an emulsifying effect on the oil. This emulsion may collect in certain portions of the system in small quantities. In the selection of the oil for the ammonia compression system, attention should be directed more particularly to the physical properties rather than the chemical properties of the oil. In a similar manner, carbon dioxide does not seem to have any particular effect upon the petroleum lubricating oil, under which condition it will be observed that the lubrication of carbon dioxide compressors is quite similar to that of the ammonia compressor.

Oil for Steam Engine Cylinders.—In ice making plants, where the exhaust from the main steam engine is condensed for the purpose of supplying water for ice making as in the distilled water ice making system, especial attention must be given to the selection of the lubricating oil for the steam cylinder. Some of the cylinder oil used for the lubrication of the steam engine cylinder may find its way into the ice. The presence of the oil is always objectionable, due to the fact that it may cause a discoloration of the ice, or give it an unpleasant odor or taste. The compounded oils have been found to give more trouble in this respect than the straight mineral oil; therefore, the cylinder oils for use in the steam cylinder of ice making plants should be pure mineral uncompounded oils.

The pure mineral oil is more easily separated from the exhaust steam. In the use of compounded oils, the presence of the various constituents aids the formation of emulsion which makes the oil difficult to be removed from the exhaust and condensed steam resulting in contamination of the boiler feed or ice water.

QUESTIONS ON CHAPTER XIX.

1. What is the main principle of good packing?

2. Name four requirements of a good packing.

3. What are the different kinds of ammonia packings which are used at present?

4. Describe the types and individual characteristics of the different kinds of packing.

5. Explain where the different types of packing are used generally.

6. What are some of the general conditions in machinery which determine the kind of oil to be used?

7. What kind of oil should be used generally in a refrigerating plant?

8. Describe the methods of testing lubricating oils.

9. State the general characteristics of oil suitable for compressors.

10. What kind of oil should be used in steam engine cylinders in refrigerating plants?

CHAPTER XX.

WORKING TEMPERATURES AND PRESSURES.*

Working Temperatures and Pressures.—In thermodynamic apparatus, certain working fluids must be employed in order for the apparatus to accomplish its functions. The physical properties of the various working fluids are characteristic of each fluid, and in general the pressures and temperatures in the thermodynamic apparatus are the physical properties that determine the selection or rejection of a given substance. As indicated in Chapter III, the relation between the temperature and pressure of a given working fluid is always definite; that is, increasing the temperature will cause an increase of pressure, while a decrease of temperature will cause a decrease of pressure.

In certain industrial processes the magnitude of the pressure may be a determining factor in the selection of the working fluid. Thus, in a steam power plant, steam is used in order that the work may be performed by allowing the steam to expand or exert its pressure against a movable piston.

On the other hand, the industrial processes may be such that the temperatures are the determining factors. For instance, in an ice making plant it is desired to remove the heat from brine and maintain the brine at a temperature of 15°. In order to do this some substance must be placed near the brine which has a lower temperature. Thus, in all industrial plants employing thermodynamic apparatus, certain working temperatures and pressures of the fluid must exist according to the nature of the plant.

Temperatures and Refrigeration.—The science of refrigeration deals with the process of cooling bodies or substances below the temperatures of the surroundings by the extraction of heat from the body or substance in question. Thus, in the ultimate analysis, it should be

* Considerable matter in this chapter has appeared in articles written by the author for *Power* at various times, and permission has been granted for its re-use here.

noted that refrigeration is a thermal process in which temperatures are of primary importance. It should be further noted that the process of cooling of a body implies the lowering of the temperature of the body.

One of the fundamental laws of physical nature states that heat always tends to flow or pass from a body or substance at a higher temperature to a body of lower temperature. Thus, if it is desired to hold a cold storage room at 25° F., it is necessary to place in the room a material that has a lower temperature than the room itself.

Hence, a pipe-coil containing ammonia or any other refrigerant is placed in the room. The pressure is maintained in the coil so that the temperature of evaporation may be 0° F. Then, due to the natural tendency, the heat in the rooms tends to flow through the coil into the boiling ammonia. The total amount of heat that may be transferred under these conditions depends upon the mean temperature difference, the amount and arrangement of the cooling coil surface, and the unit heat transfer coefficient of the coil surface. It is obvious that the mean temperature difference is of primary importance.

Temperature Differences in Cold Rooms, Etc.—The magnitude of the temperature differences between the various elements of the refrigerating plants depends upon economic considerations. Thus, the cost of power and the cost of the pipe-coil determine, to a very large extent, the magnitude of these temperature differences.

The relationship between the temperature difference, the amount of heat transmitting surface, and the quantity of heat transfer are shown by the following formula:

$$H = K \times A \times \text{m.t.d.}$$

In the foregoing formula, m.t.d. represents the mean temperature difference between the substances. If the temperatures of either of the substances vary, the mean temperature difference, m.t.d., may be determined by the methods given in Chapter VIII. From the Chapter VIII it will be remembered that the mean temperature difference is equivalent to the greater temperature difference of the two fluids multiplied by a coefficient. Expressed in symbols, this may be stated as follows:

$$\text{m.t.d.} = \text{g.t.d.} \times M$$

where m.t.d. = mean temperature difference
g.t.d. = greater temperature difference
M = a coefficient

The value of the coefficient, M, depends upon the quotient of the smaller temperature differences divided by the larger temperature dif-

ference, and may be taken from Table 59 of Chapter VIII, or from Fig. 196 of this Chapter. When the temperatures of the fluids are constant, the mean temperature difference is simply the arithmetical difference of the two temperatures, and the heat that will be transferred under this condition may be expressed in symbols as follows:

$$H = K \times A \times (t_1 - t_2)$$

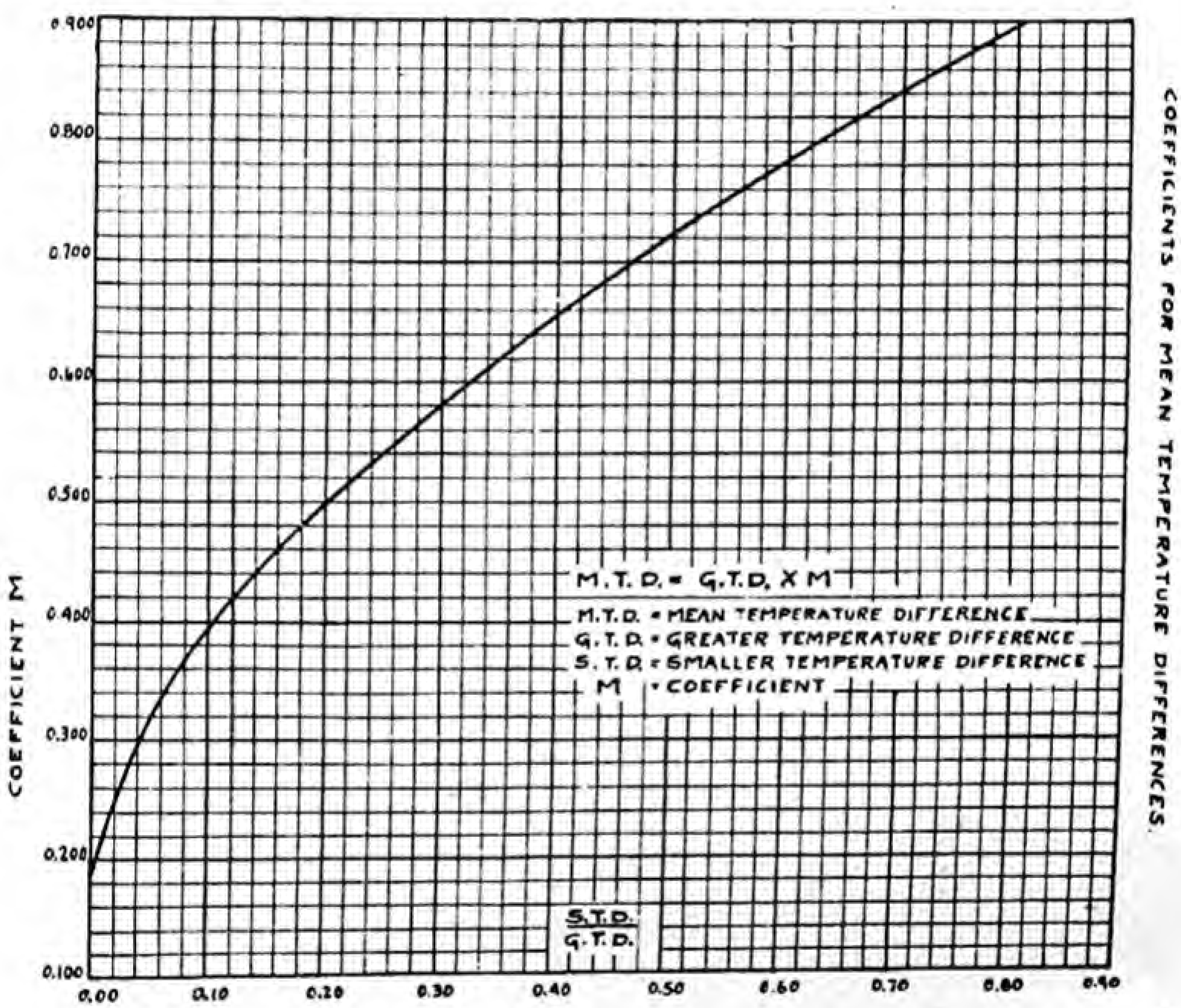

Fig. 196.—Coefficient for Mean Temperature Difference.

From the foregoing, for a given amount of surface, it will be observed that the heat transfer is directly proportional to the mean temperature difference. In other words, doubling the temperature difference would double the amount of heat transferred by the surface. Also, by means of the foregoing formula, it will be noted that if the quantity of heat to be transferred is held constant, increasing the amount of surface will affect the magnitude of the temperature difference in an inverse proportion. Thus, if the surface is doubled, the temperature difference would be decreased by fifty per cent.

Thus, it may be said that the larger amount of coil surface will produce more economical operating conditions, due to the fact that the suction pressure may be carried at a higher point. The suction pressure should be carried as high as possible and still maintain the desired

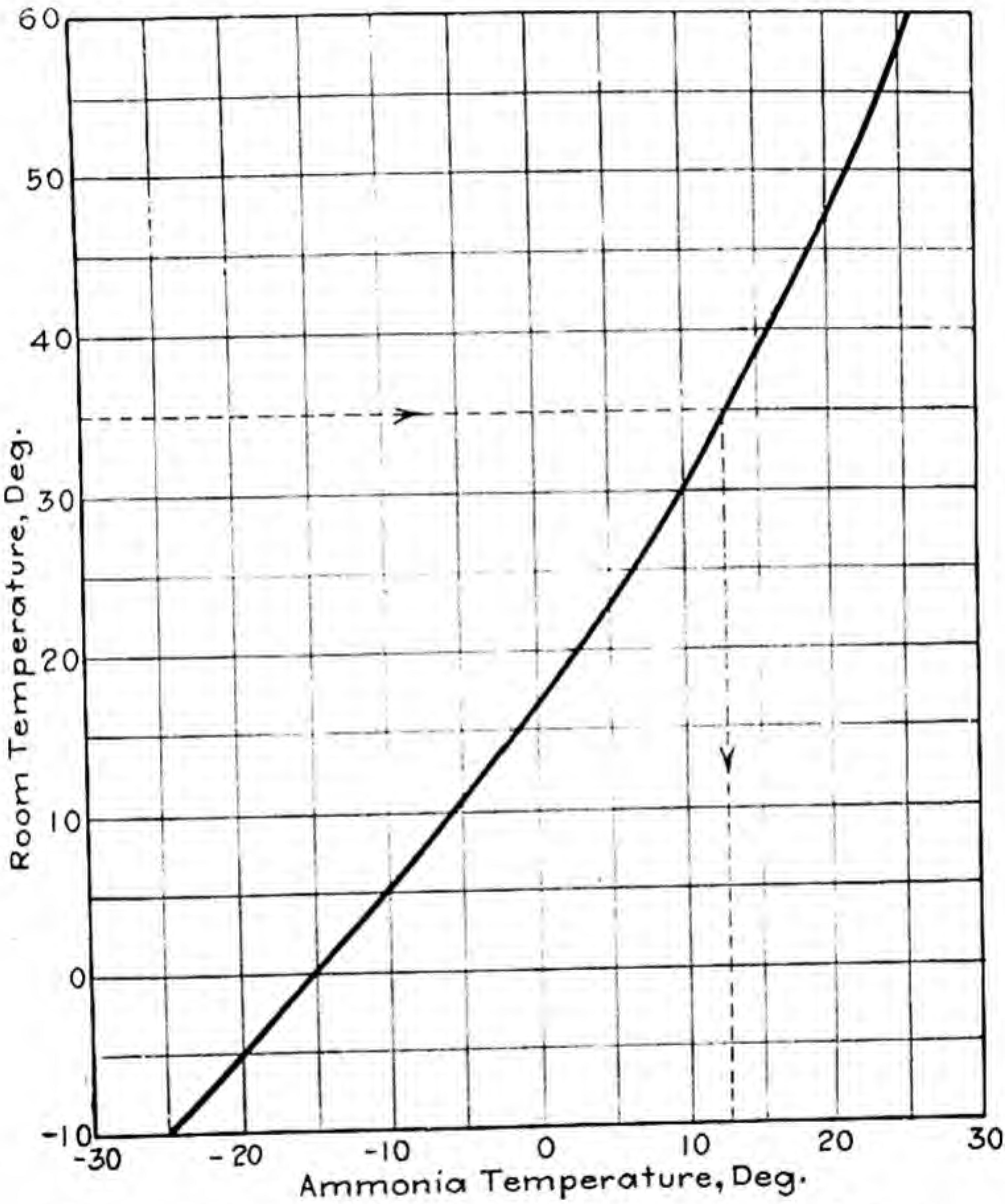

Fig. 197.—Difference of Temperature for Ammonia and Rooms Using Direct Expansion Coils.

temperatures. The principal advantage of using a higher suction is due to the fact that less power is required per ton of refrigeration as the suction pressure becomes higher. An additional advantage of higher suction is that the tonnage capacity of the compressor per cubic foot of displacement increases as the suction is increased.

In order to convey to the reader an idea of the magnitude of these temperature differences, Figs. 197 and 198 have been prepared. Fig. 197 indicates the relation between the temperature of cold storage rooms and the temperature of the evaporating ammonia in the cooling coils. These temperature differences should be maintained in order to secure economical operation.

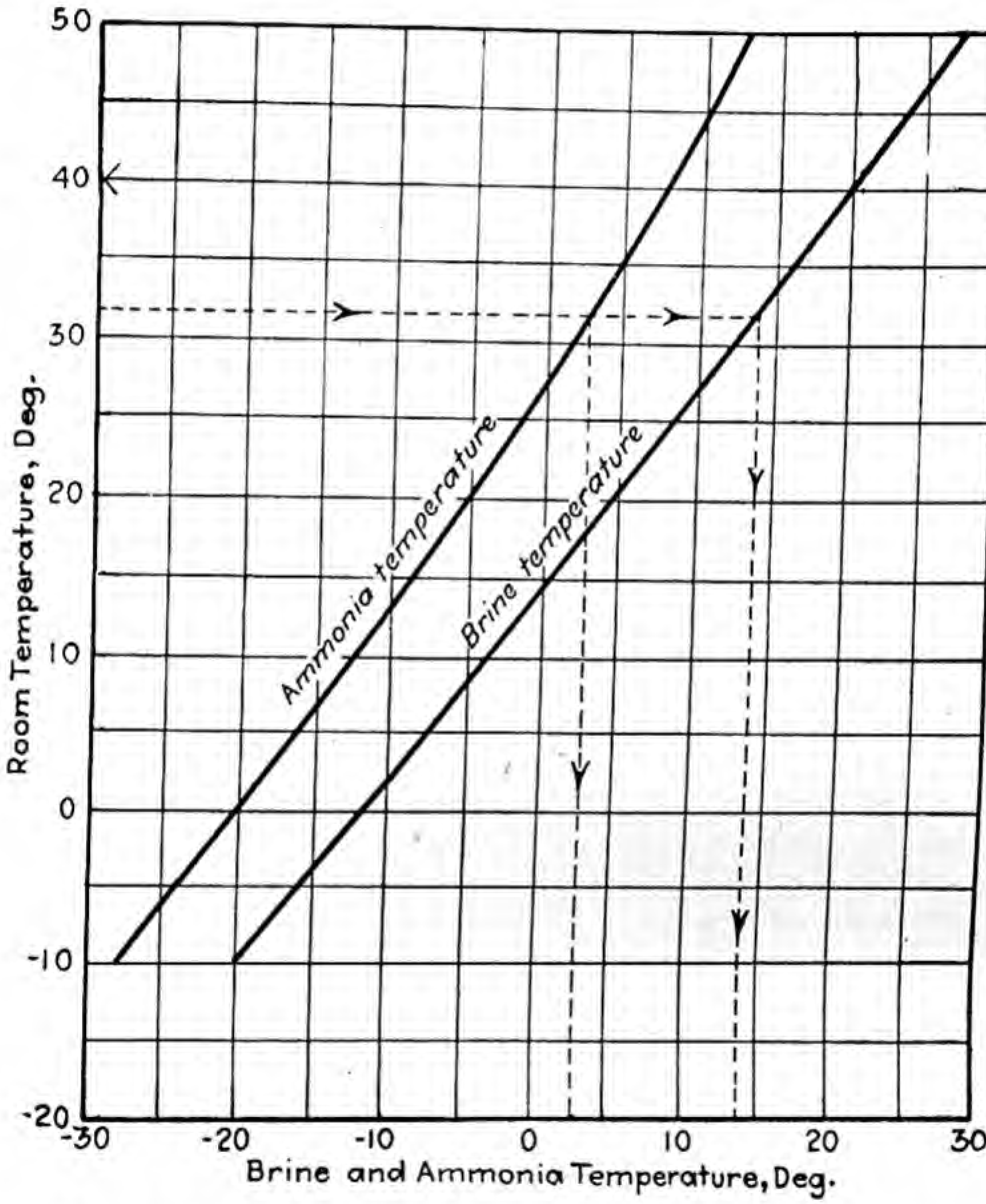

Fig. 198.—Difference of Temperatures Using Brine Coils.

The following values, taken from Fig. 197, will indicate the variation of the temperature differences as the room temperature increases:

Room temp.	–10	0	10	20	30	40	50	60° F.
Ammonia temp.	–25	–15	–5	3	10	16	22	26° F.
Temp. diff.	–15	15	15	17	20	24	28	34° F.

In a similar manner, Fig. 198 illustrates the relationship of the room temperatures, brine temperatures and ammonia temperatures, in a refrigerating plant in which the rooms are cooled by brine coils. In this type of plant the total temperature difference between the room and the ammonia is somewhat greater, due to the fact that the brine must be at a lower temperature than the room, and in turn the ammonia temperature must be below that of the brine in order to remove the heat.

The following tables will show the temperature differences which will promote efficient operation:

Room temperature, deg. F.......	-10	0	10	20	30	40	50
Brine temperature, deg. F.......	-20	-12	-4	4	12	20	28
Temperature difference, deg. F..	10	12	14	16	18	20	22

Room temperature, deg. F.......	-10	0	10	20	30	40	50
Ammonia temperature, deg. F....	-28	-20	-13	-6	1	8	13
Temperature difference, deg. F..	18	20	23	26	29	32	37

Brine temperature, deg. F.......	-20	-12	-4	4	12	20	28
Ammonia temperature, deg. F....	-28	-20	-13	-6	1	8	13
Temperature difference, deg. F...	8	8	9	10	11	12	15

By reference to Figs. 197 and 198 the temperature of the ammonia that will produce the desirable results may be ascertained. Then by reference to tables of properties of ammonia, the pressures of saturated ammonia corresponding to various temperatures and hence the suction pressures that should be used may be obtained.

Suction Pressures and Temperatures.—From the foregoing, it is possible to obtain the suction pressure and temperature that should be used to accomplish the desired result. It is of interest to note the ways and means of securing these temperatures and pressures, as well as the maintaining of same.

Since it is the province of refrigeration to deal primarily with temperatures, it is obvious that efficient methods of temperature indications must be employed in order to accomplish the proper results.

In the first place, the thermometers should be accurate temperature indicators; they should possess extreme sensitiveness and they should be rugged in strength and of durable construction. In the second place they should be installed properly. They should be located so that they may be easily consulted. This condition will result in intelligent readings at the proper intervals.

In case of the suction side of the system, the thermometers are installed near the coil ends, or, preferably, in the suction connection near the compressor. Of course, the type and size of plant governs the amount and kind of thermometers and gauges which should be installed. Generally speaking, the investment of capital in such instruments is the wise and proper plan.

Ordinarily, on suction connections to compressors, a recording pressure gauge, together with a thermometer, will afford excellent means of regulating the suction pressure. On the other hand, the advantages of having permanent records of pressures and temperatures in the form of charts from recording instruments are obvious. The plant may be operated efficiently only when the knowledge of the conditions is accurate.

After the temperature of the return gas has been ascertained, the exact state of the refrigerant may be determined by consulting a table of the properties of the refrigerant.

Furthermore, the actual regulating of the evaporating coil pressure and temperature is obtained by the proper manipulation of the "expansion" valves. These valves should be regulated so that the suction gases will have only 5° to 10° of superheat at the compressor. The ease of regulation is affected by the design of the piping system, the installation of traps and separators, etc.

When the vapors come to the compressor in the saturated state the largest weight of refrigerant will be handled and thus the larger amount of refrigeration will be produced. In the case that horizontal compressors are used, the frost should be allowed to come back to the compressor manifold. Likewise, in some plants where the suction pressure is quite low, it may be found advisable to allow a small amount of liquid to enter the compressor to provide a means of preventing high temperatures at the end of compression.

In a similar manner, the use of thermometers facilitates the operation of vertical compressors, since in this type of compressor it is not so advisable to allow liquid refrigerant to enter the cylinder. This is due to the fact that the vertical machine may not handle the moist vapor as efficiently as the horizontal type. The use of thermometers will show the proper degree of superheat for proper operation.

Many operating problems may be readily solved by reference to accurate temperature indications. The case of exceedingly high suction gas temperature may be considered. In a plant where the suction gas is many degrees above the temperature corresponding to the pressure, several causes may be suspected, such as leakage of compressor valves or lack of ammonia.

In addition to the variation of the temperatures in the low pressure side of the refrigerating system, it may be said that the pressures will vary considerably. The suction pressure may be either too low or too high for a given set of working conditions. There are many things that may cause an untrue pressure. One of the things that may commonly occur in ice making and refrigerating plants is the clogging of the expansion valve either by frozen water or oil, scale, or other foreign matter. Another thing that may happen sometimes in a refrigerating

plant is that the suction strainer, which is generally located just before the compressor, will become clogged in practically the same manner as the expansion valve. In either case, it requires an increased pressure to force the refrigerant through the obstructed areas, which means that the pressure in the coils or in the connections near the compressor will be materially reduced.

The collection of foreign matter on the evaporator surface adds resistance to the flow of heat. As the quantity of heat to be transferred and the magnitude of the surface remains constant, the heat can be made to flow against the increased resistance only by increasing the temperature difference. This, in turn, will mean a lower temperature of the refrigerant, which means that the suction pressure must be carried at a correspondingly lower point. The lack of a sufficient quantity of refrigerant will sometimes cause a low suction pressure. This is probably due to the fact that when the coil is normally charged with liquid, the presence of the liquid refrigerant will tend to increase the rate of transmission of heat.

In a similar manner, a number of things may cause a high suction pressure. Sometimes, when the expansion valve has been clogged, the pressure in the evaporator will become so reduced that the obstruction will be forced through the restricted area in the expansion valve. Immediately following this action, an undue quantity of liquid refrigerant passes into the evaporator which causes the suction pressure to increase above the normal. If there is not sufficient quantity of liquid in the refrigerating system to properly seal the outlet in the liquid receiver, the gaseous refrigerant will pass directly through the evaporator. Under continued operation in this manner, the suction pressure can become higher than the normal pressure. If the suction valve on the compressor leaks considerably, the compressor will not be able to remove the vapors of the refrigerant as fast as they form, under which condition the suction pressure will go to a higher point, the frost will melt from the suction connections, etc.

Discharge Temperatures.—The proper regulation of the discharge temperature is an important consideration in the successful operation of the refrigeration plant. This is due to the fact that if the temperatures at the end of compression are allowed to become fairly high, the ammonia or other refrigerant may tend to disintegrate into the elementary constituents. This leads to the presence of non-condensable or permanent gases in the system.

The temperature of the gases at the end of compression depends upon the ratio of the absolute condenser pressure to the absolute suction pressure, or in other words, the ratio of compression, and upon the degree of superheat of the suction gases. The degree of superheat

or the condition of the suction vapors may be regulated by means of proper adjustment of the expansion valves.

Under ordinary conditions, it is advisable to provide means of ascertaining the temperature of the discharge gases. Indicating or recording thermometers may be installed. However, it should be evident that the operation of the compressor is determined primarily by the temperature and pressure of the suction vapor, and that the discharge temperature will take care of itself if the conditions are what they should be on the suction side of the machine.

The thermometer in the discharge main will be found useful in the event that the thermometer in the suction main is broken or out of order, since it would be possible, by reference to the engine room log, to determine just what the suction temperature should be in reference to the discharge temperature.

To sum up the considerations of the temperatures and pressures of the suction and discharge gases, the following statements may be made: The suction pressure should be carried as high as possible, maintaining at the same time the desired temperatures. The suction vapor should enter the compressor as near as possible to the saturation temperature corresponding to the pressure, without increasing the danger of getting liquid ammonia into the compressor cylinder; that is, the suction vapor should be only slightly superheated. The discharge pressure should be kept as low as possible, giving especial attention to the quantity of water available and the amount of condenser surface. The discharge temperature should be kept as low as possible.

Thus, the operating engineer must be provided with means of ascertaining the temperature and pressure of the refrigerant as it passes through the compressor. The engineer will then be in a more advantageous position to operate the compressor in the most efficient manner.

It is well to observe some of the things which will cause the compressor to become extraordinarily hot, as it may at times. If the various parts of the compressor are not accurately machined and in proper alignment, undue amount of friction may result. This is especially true if the center line of the cylinder does not coincide exactly with the center line of the frame. Another action which tends to heat up the compressor cylinder is the leakage of the compressed gases around the piston rings or through the discharge valve. This produces an action which is similar to churning and the temperature will gradually increase. Insufficient lubrication of the compressor cylinder and the compressor stuffing box will allow the friction to increase to an excessive amount. The work required to overcome the friction is converted into heat which heats the compressor several degrees above normal operating temperature.

Other Uses of Thermometers.—The use of thermometers is essential to the efficient operation of ammonia condensers. The performance of the various sections may be controlled by observing the temperature of the water leaving the various coils. These temperatures may be determined by installing suitable thermometers.

The temperature should be approximately the same for all of the coils to operate evenly. By proper temperature indications and manipulation of valves, it is an easy matter to operate the various types of condensers. Of course, it is well to employ thermometers to indicate the temperature of the water into and out of the condenser.

The same general considerations apply to the operation of brine coolers also. In the event that more than one cooler is employed, thermometers should be located in the brine outlets of each coil or cooler. This facilitates the dividing of the cooling work evenly among the coolers. Of course, thermometers should be installed in the main brine lines to indicate the average cooling of the total amount of brine.

It is well to use recording thermometers for determining the temperature range of the water through the condensers and the brine through the brine coolers. These furnish permanent records of the daily operating conditions.

Since the temperature of the liquid ammonia just before the expansion valve has a direct bearing upon the refrigerating effect of the ammonia in the evaporating coils, it is evident that a thermometer should be installed in the liquid line at a point just before the expansion valve. The temperature at this point has not only a bearing on the refrigerating effect, but also is a factor in determining the advisability of aftercooling the liquid ammonia.

It is hardly necessary to mention the use of indicating and recording thermometers for ascertaining and recording the temperatures of cold storage rooms and other refrigerated spaces.

The operating engineer should study his own individual plant with the view of determining the most economical working temperatures and pressures. He should then adopt efficient means of ascertaining and maintaining these temperatures and pressures.

A Guide to Correct Compressor Operation.*—Leakage in a compressor might go on for years running up the power bills and cutting down production, and not be detected by the engineer. But it is a simple matter to have a constant indication of the correct performance of the machine. Fit a thermometer well in the discharge of the compressor, and it will tell, with the aid of the chart given in Fig. 199, just how good or how bad the performance is that day. At the same time

* Thomas M. Gunn, Vacuum Oil Co., in N. A. P. R. E. Bulletin No. 40.

another thermometer should be installed on the compressor suction pipe, if one is not already there, in order to check up on the condition of the suction gas.

Here are the reasons why the thermometer on the discharge tells such an important story. The temperature of discharge gas is high because each pound of gas compressed has received added heat energy just equal to the amount of work expended in its compression. If

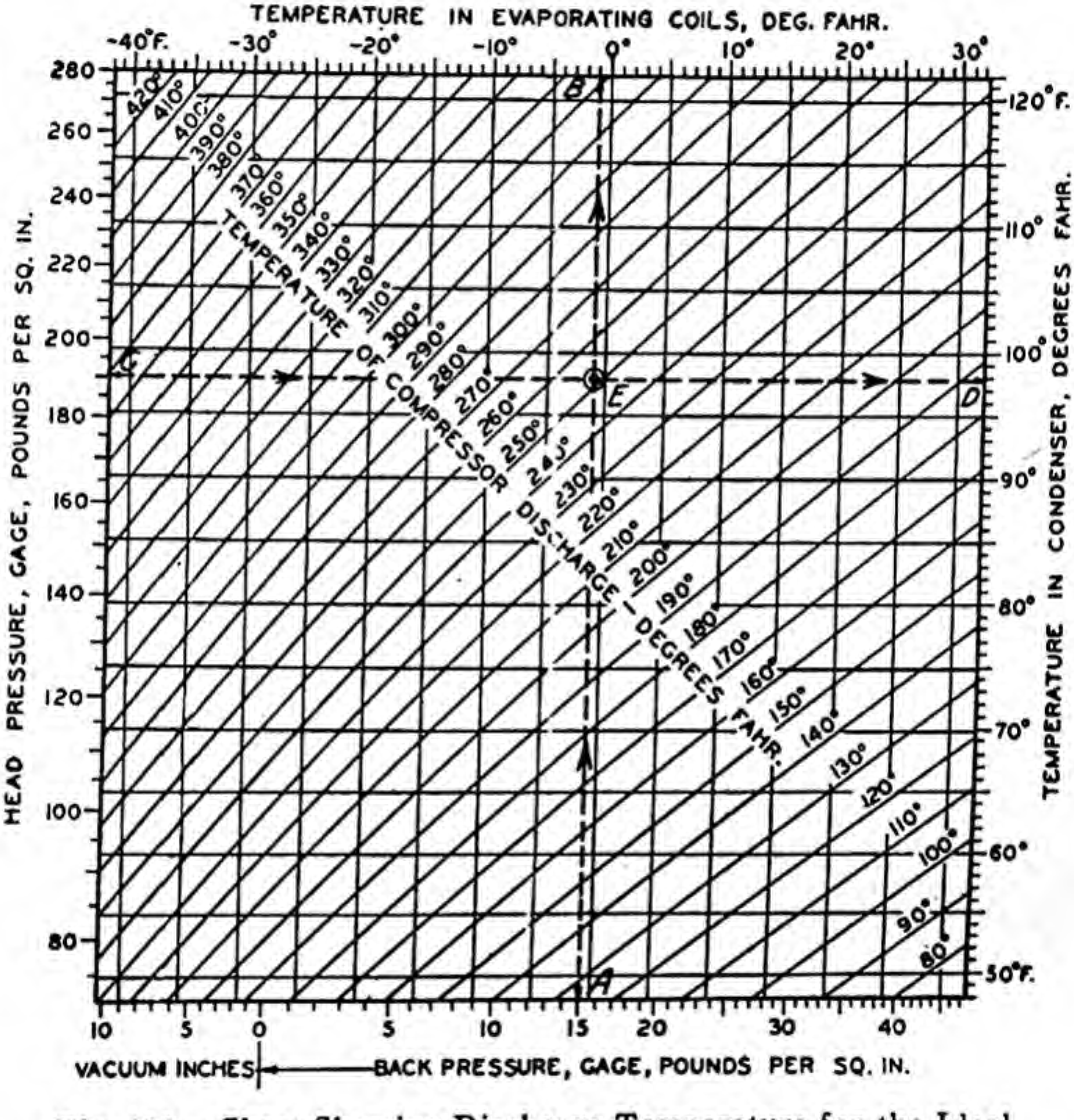

Fig. 199.—Chart Showing Discharge Temperature for the Ideal Ammonia Compressor.

compression is efficient, the work is comparatively small. There is a correspondingly small rise of temperature during compression. If compression is less efficient, there is more work done on each pound of gas, and its temperature is correspondingly greater. Each of these causes inefficient performance, tends to increase the work done per pound of gas compressed, and hence to increase the discharge temperature.

The diagram shown in Fig. 199 gives the discharge temperature in an ideal compressor, such as was described at the beginning of this article. Fortunately, in an actual compressor operating properly and normally, the discharge temperature is usually not far from what is shown by this chart. All compressors have their peculiarities, and each will show certain tendencies to give higher, or possibly lower, discharge temperatures than given by this chart. A change, such as a broken piston ring or chipped valve, will have an immediate effect on this discharge temperature.

As an example of the use of this chart, suppose that we read gauges and find the head pressure to be 190 lbs. and the back pressure to be 15 lbs.; and that the thermometer on the discharge reads 290° F. while that on the suction reads 5° F.

Referring to the chart, Fig. 199, find the point at the bottom for 15 lbs. back pressure, and from there draw a vertical line *A-B* to the top scale, where the temperature of evaporation may be read, which is "minus one degree F." Hence the suction gas, as shown by the thermometer, is superheated 6° F.

Next, find the head pressure, 190 lbs. on the scale at the left, and draw the line *C-D* horizontally across the diagram. The point *D* on the scale at the right shows a condensing temperature of 90° F. Where the two lines intersect, *E*, notice the diagonal lines. *E* is between the two lines marked 250° and 260°, and may be read 252° F. This is the discharge temperature of the ideal compressor. This particular compressor, we have assumed, showed a discharge temperature of 290° F., or 38° higher than the figure for the ideal compressor.

This difference is more than there should be in a well designed compressor. If we had found a suction superheat of 20° or 25°, instead of only 6°, it would be sufficient to explain this high discharge temperature, but as it is, there is probably some other cause of high discharge temperature, such as:

Valves stick open or stick shut
Valve springs too heavy
Valve lift insufficient
Valves or seats injured and leaking
Piston rings broken
Piston rings so worn as to fail to press on walls
Ring slots in piston worn
Ring ends poorly fitted
Rings stuck in slots by deposits
No effective oil seal between rings and cylinder walls, producing leakage and frictional heat.

When the engineer knows there is trouble to be remedied, he soon finds it. This chart and the useful thermometer soon become infallible

indications in the hands of a man who watches conditions from day to day—especially if he keeps a log.

There are, of course, conditions that may reduce the discharge temperature below that of the chart. For instance, the expansion valve may be so open as to cause some liquid to return to the compressor, perhaps as a spray or fog. This has a marked lowering effect on the discharge temperature.

Engineers in certain parts of the world like to have such a liquid fog in the suction gas, and even go so far as to fit a small expansion valve to inject into the suction. It reduces the temperature of the cylinder and stuffing box, and so makes lubrication easier. Where this practice is followed it is common to regulate the liquid injection in accordance with the discharge of temperature. This has the objection, however, that with "wet compression," as this is called, there is no chance to detect leakage or other faults, as we have explained, by means of the discharge temperature, because any of these smaller effects are covered up by the greater effect of the liquid injection.

All of the foregoing discussion has referred to ammonia compressors. Carbon dioxide compressors are often installed in hospitals, theatres and other public buildings. The same condition holds for them also, except that another chart is necessary. The data of carbon dioxide is not as exact as for ammonia, but the accompanying chart, Fig. 200, has been compared with practice and appears to have equal value for use in checking the performance of these compressors.

Pressures in the Compressor Cylinder.—In order to operate the compressors in the most efficient manner, the operator must know how the pressures vary during the strokes of the piston in the compressor cylinder. The operator generally makes use of a pressure indicator for determining the magnitude of the pressures at the different points of the stroke. There are several efficient types of indicators upon the market, any of which if properly handled will provide a means of ascertaining the pressures at the different points of the stroke.

The general method of the variation of the pressures in a compressor cylinder during a revolution of the compressor may be observed by an inspection of Fig. 201. This figure shows a cross section of a compressor cylinder upon which the diagram *CDEF* has been superimposed. The distances along the line *OB* represent units of pressure, while the distances along the line *OA* represent the different magnitudes of volumes. If the piston is at the right-hand end of the cylinder and the pressure is 30 lbs. per sq. in. abs., this condition would be represented by point *C* on the diagram. As the piston travels to the right-hand end of the cylinder the gas is compressed until at point *D*

it is equivalent to or slightly higher than the pressure in the condenser. The discharge valve will, therefore, open and the compressed gas will be discharged into the condenser as indicated by line *DE*.

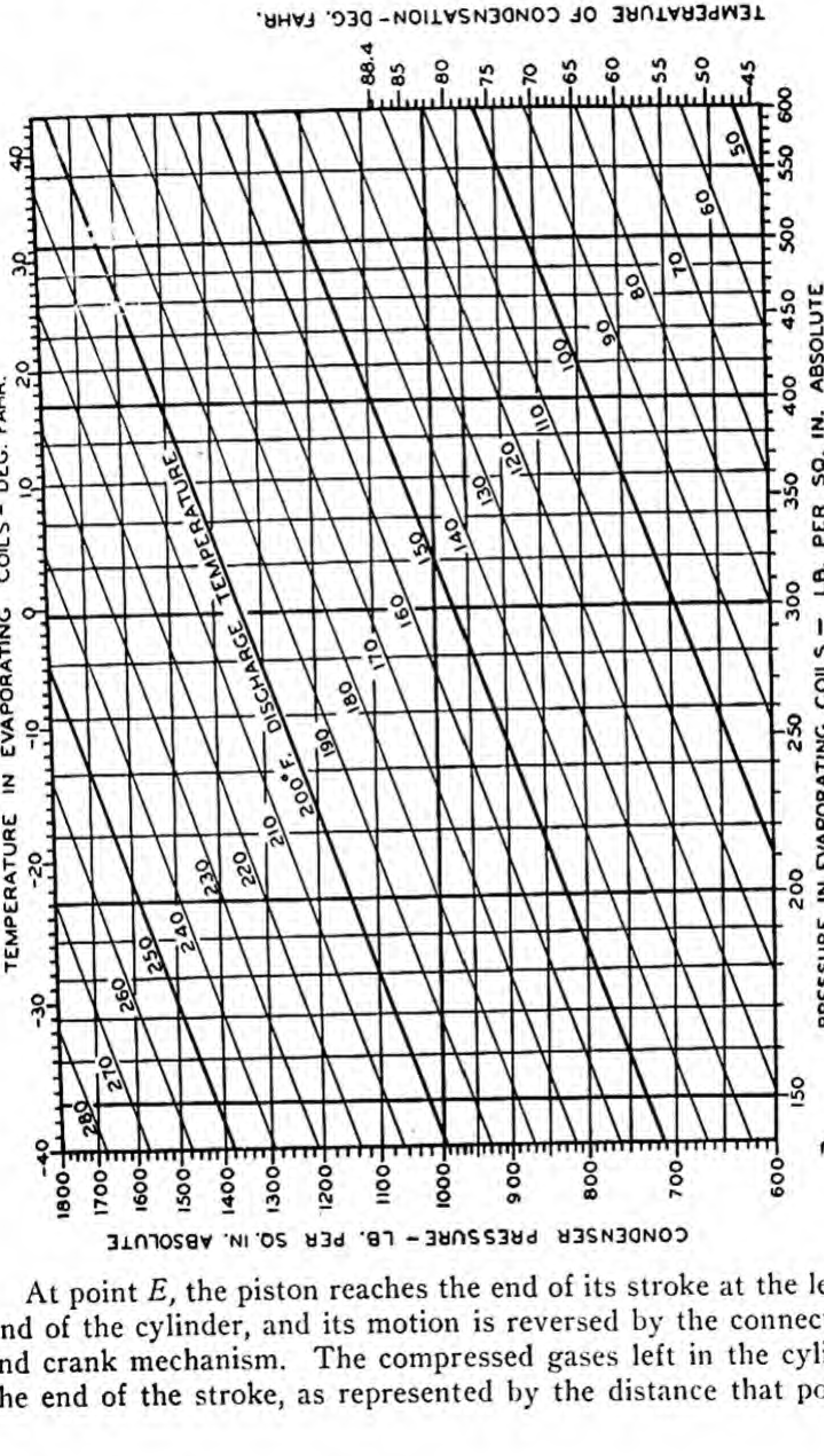

Fig. 200.—Chart Showing Discharge Temperature of Carbon Dioxide Compressor Under Ideal Conditions.

At point *E*, the piston reaches the end of its stroke at the left-hand end of the cylinder, and its motion is reversed by the connecting rod and crank mechanism. The compressed gases left in the cylinder at the end of the stroke, as represented by the distance that point *E* is

from the cylinder head, will expand as the piston moves toward the right, until at point *F* the pressure has been reduced within the cylinder to or slightly below that in the evaporator. At this point *F*, the suction valve opens and the vapor of the refrigerant flows into the cylinder as indicated by line *FC*, until the piston comes to rest at the right-hand end of the cylinder. The diagram *CDEF* therefore represents graphically the variation of the pressures in the cylinder during all points of the suction and compression strokes.

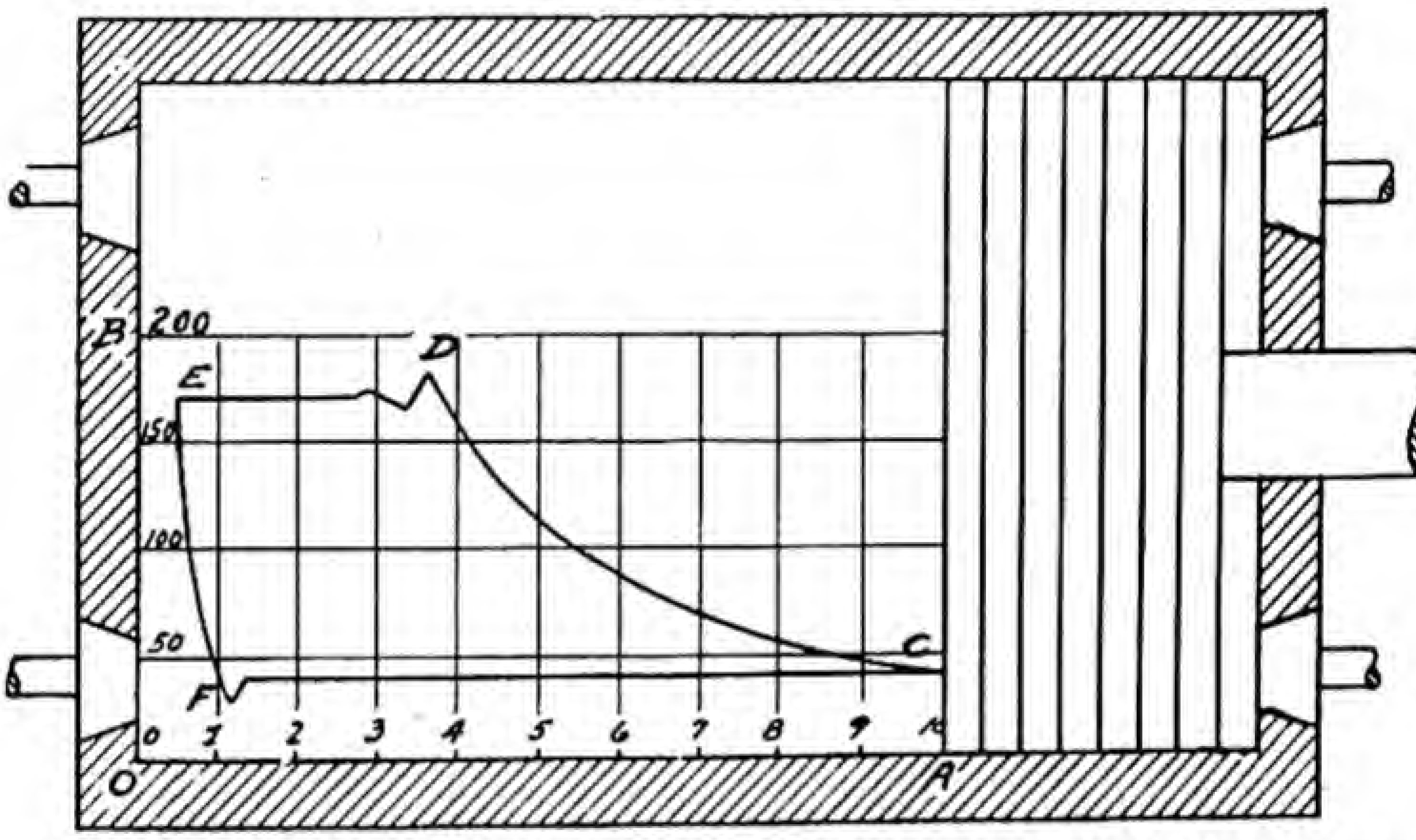

Fig. 201.—Compressor Cylinder with Pressure Volume-Diagram.

The pressure indicator is simply a mechanical device for drawing to a reduced scale the diagram *CDEF* of Fig. 201. The stroke of the piston is generally reduced to 3 or 4 in., while the height of the diagram is reduced to 1½ or 2 in. by using a suitable scale for the spring in the indicator. From Fig. 201 it will be noted that the distance *OA* represents volume or space, while the distance *OB* represents force or pressure. The diagram *CDEF*, therefore, really represents the relationship of the pressures and volumes of the ammonia during the cycle of operation.

Relationship of Pressure and Volume.—The relation between the pressure, volume and temperature of any perfect gas is definite and is characteristic for each substance. Boyle's law states that if a gas is compressed at a constant temperature the pressure will vary inversely as the volume. This means that the product of the pressure and volume at the various points of the compression will remain constant. Thus, if P_1V_1, P_2V_2, P_3V_3 represent the various conditions the following will be true:

$$P_1V_1 = P_2V_2 = P_3V_3 = \text{etc.}$$

When a gas is compressed or expanded at a constant temperature, the process is termed "isothermal compression or expansion."

In the study of the actions of vapors and gases during compression and expansion it will be found that the various gases will obey certain characteristic laws which may be expressed symbolically as follows:

$$PV^n = \text{a constant}$$
P = absolute pressure, lbs. per sq. ft.
V = volume of 1 lb.
n = characteristic index of exponent

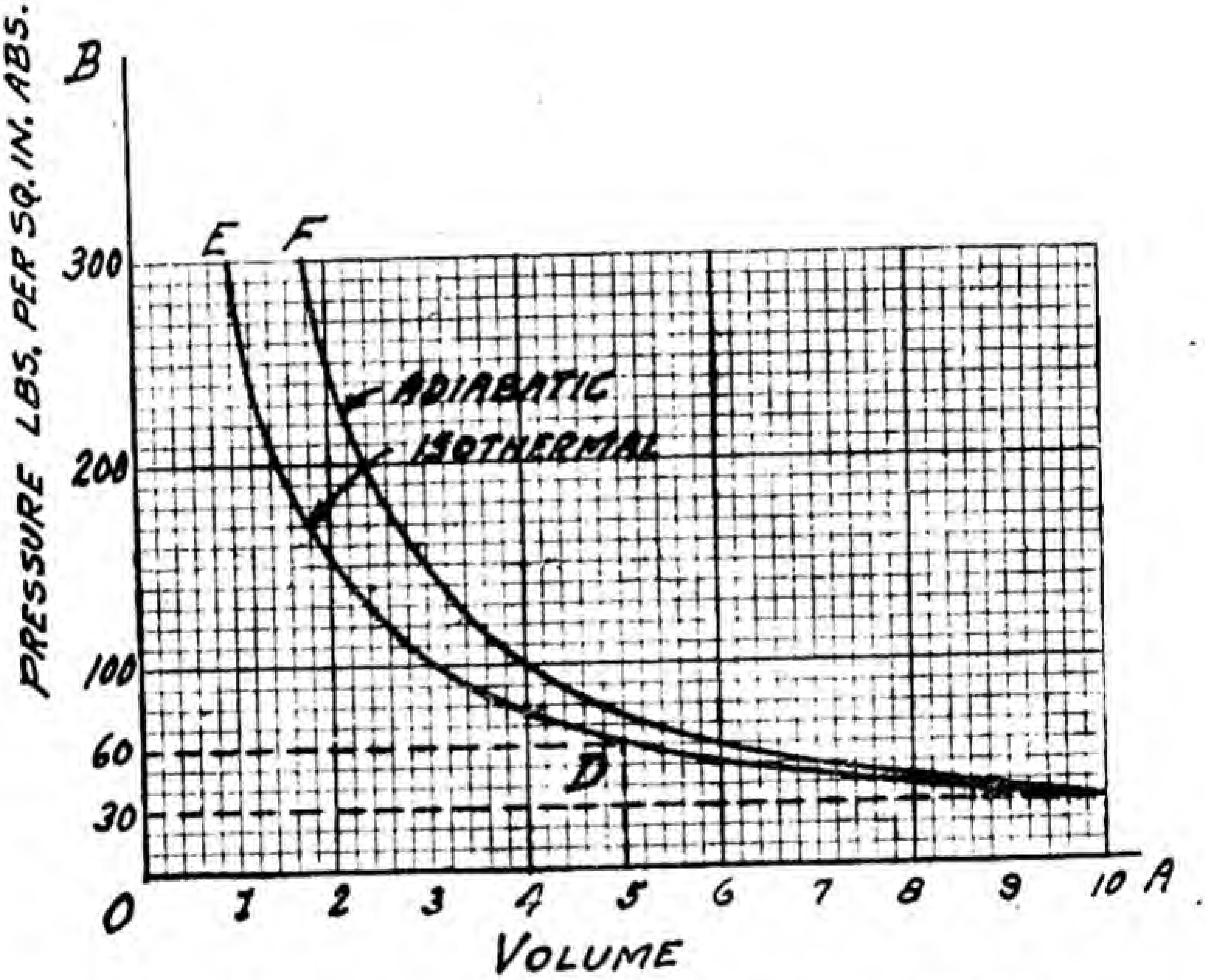

Fig. 202.—Isothermal and Adiabatic Compression Curves.

The foregoing law means simply that the product of the pressure in pounds per square foot and the volume raised to the power indicated by the exponent will always equal a certain constant for a given gas. If the mode of the compression of the gas is such that the substance neither receives nor rejects heat as such, the process is said to be adiabatic. In this case the value within is determined by the magnitude of the specific heat at constant pressure and at constant volume. The exponent is generally denoted by k, and the relationship is shown as follows:

$$n = k = \frac{C_p}{C_v}$$

where k = characteristic exponent for adiabatic compression
C_v = specific heat at constant pressure
C_p = specific heat at constant volume

The numerical values of C_p and C_v for ammonia are such that the value may be represented by 1.28. The foregoing law, therefore, means that the product of the pressure times the volume raised to the 1.28 power at a particular stage in the compression will be equal to the pressure times the volume to the 1.28 power at any other stage. This may be represented symbolically as follows:

$$P_1V_1^{1.28} = P_2V_2^{1.28} = P_3V_3^{1.28} \text{ etc.}$$

Isothermal and Adiabatic Compression.—The foregoing laws determine the relationship between the pressures and volumes of ammonia or any other refrigerant during compression. When the compression is isothermal the temperature and also the product of the pressure times the volume are constant. This relationship may be more thoroughly appreciated by referring to the pressure-volume diagram shown in Fig. 202. In this diagram the volumes are represented by the distances along line *OA*, while the pressures are represented by the distances along line *OB*. Thus, if the pressure of a gas at point *C* is known to be 30 lbs. per sq. in. abs. and the volume is known to be 10 or unity, the volume and pressure at any other point may be readily determined. Thus, at point *D*, the gas may have been compressed at a constant temperature until the volume has been reduced by one-half, or in other words, point *D* is vertically above 5 on the axis *OA*. By means of Boyle's law the following equation may be written:

$$P_cV_c = P_dV_d$$

Since the volume at *C* is equal to 10, the volume at *D* is equal to 5, and the pressure at *C* is equal to 30, the pressure at *D* may be found as follows:

$$P_d \times 5 = 30 \times 10$$
$$P_d = 60$$

The foregoing law may be used to calculate the pressures at any point along the isothermal compression curve, when the conditions at one particular point are known.

The determination of the location of the adiabatic compression curve from the perfect gas compression law (PV^n = a constant) is subjected to a slight inaccuracy due to the fact that the exponent "n" is not strictly a constant for ammonia. In fact, the value of "n" will vary from 1.25 to 1.30, depending upon the conditions of the gas com-

pression. When the adiabatic compression law PV^n = a constant is used it is advisable to use a value of n = 1.28, as representing the average for usual conditions. This method of locating the adiabatic compression curve in this manner has been described in many books and magazines so that a repetition of the same is not necessary here.

A more modern and accurate method of locating the theoretical adiabatic compression has been developed by the author (Feb., 1926) in some unpublished research work. In this new method, direct use is made of the properties of superheated ammonia, as given in the United States Bureau of Standards Ammonia Tables (Pages 71-84).

During the adiabatic compression of the ammonia, heat, in the form of heat, is neither absorbed nor rejected by the gas. Remembering that entropy may be defined as the heat added or rejected during a process divided by the absolute temperature, it will therefore be observed that the entropy during adiabatic compression must remain constant. This fact, together with data given in the Bureau of Standards ammonia tables, make it possible to locate the adiabatic compression curve without reference to the perfect gas law.

The operation of this method of locating the adiabatic curve is best illustrated by the following example: It is assumed that the pressure at the beginning of the compression is 35 lbs. per sq. in. abs. and that the temperature of the ammonia vapor is 50° F.; that the compression pressure is 170 lbs. per sq. in. abs.; that the compression is adiabatic, that is, a constant entropy process. From the superheated ammonia tables it will be found that the entropy of the vapor at 35 lbs. and 50° F. is 1.3756, and the specific volume is 8.895 cu. ft. per lb. After compression the pressure is 170 lbs. abs. and the entropy is 1.3756 also. From the tables it is found at this point, that the volume is 2.593 cu. ft. per lb. It may be assumed that the ammonia compressor holds just 8.895 cu. ft. at the beginning of compression. The percentage of displacement produced to compress the ammonia gas to 170 lbs. is found as follows:

$$\frac{8.895 - 2.593}{8.895} = 0.709 = 70.9\%$$

The relative volumes and corresponding displacements for adiabatic compression to various pressures may be found in a similar manner, as shown in the following tabulation:

Absolute pressure	Percentage reduction of volume
35	0.00
50	23.8
75	44.8
125	62.8
200	74.3
300	81.3

It is evident that the points thus located represent the piston movement in percentage of reduction of total inside volume of the compressor cylinder. If the cylinder has no clearance, the points so located represent percentage of reduction of the stroke volume. In a cylinder having clearance, the points so located represent percentage of reduction of stroke volume plus the clearance volume.

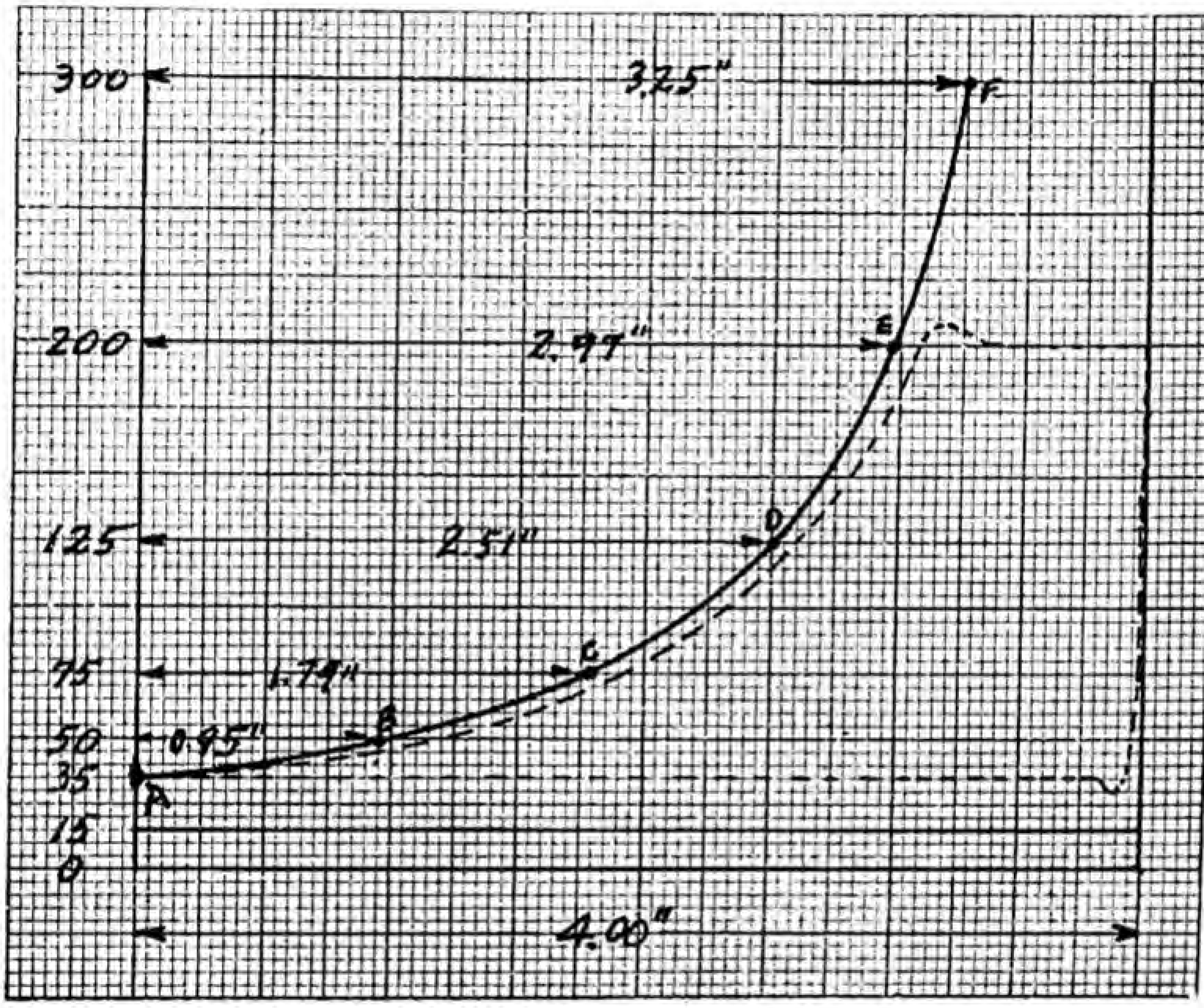

Fig. 203.—Adiabatic Curve on Indicator Card.

It will be observed that this method provides an accurate way of laying out the theoretical adiabatic compression curve. It is obvious that the method adapts itself to the determination of the location of the curve, as influenced by both temperature and pressure of the gas at the beginning of compression. Under actual conditions the ammonia vapor has several degrees of superheat at the end of the suction stroke. In this method, it is possible to consider this fact, and thus locate the compression curve more accurately.

The practical use of the method is illustrated by Fig. 203, in which it is assumed that pressure at the beginning of compression is 35 lbs. abs. and the temperature of the gas is 50° F.; that the compressor has no clearance and that the length of the indicator card is exactly 4.00

ins. and various horizontal lines are drawn on the card at 50, 75, 125, 200 and 300 lbs. abs. pressures, using, of course, the same scale as the scale of the indicator spring. The horizontal distance from a vertical line through the "toe" of the card along the 50 lb. pressure line to the adiabatic compression curve is found as follows, using the percentage of volume reduction given in a previous paragraph:

At 50 lbs. abs. compression pressure,

$$0.238 \times 4.00 = 0.95 \text{ ins.}$$

This point may therefore be located 0.95 ins. from the vertical line through the toe of the card along the 50 lbs. pressure. Additional points on other pressure lines may be found in accordance with the following tabulation:

Compression pressures	Percentage volume reduction	Length of card	Horizontal distance to point on adiabatic curve
50	0.238	4.00 ins.	0.95 ins.
75	0.448	4.00 ins.	1.79 ins.
125	0.629	4.00 ins.	2.51 ins.
200	0.743	4.00 ins.	2.97 ins.
300	0.813	4.00 ins.	3.25 ins.

These points may be set off on the card by means of a scale graduated in hundredths of an inch, by measuring along the various compression pressure lines from a vertical line through the toe of the card.

The method may be used also to lay out adiabatic compression curves on cards taken from compressors with clearance. In this case, the factors representing the percentage of volume reductions up to the various pressures are multiplied by the length of the card plus an additional amount in proportion to the amount of clearance. The horizontal lines are then laid off on the card at various pressures. A vertical line is erected through the toe of the card, as previously mentioned. The horizontal lines obtained in the foregoing manner are laid off along the horizontal pressure lines, thus locating several points on the theoretical adiabatic compression curve. A smooth curve may then be drawn through these points.

The percentages of volume reductions vary with the pressures at beginning of compression and various compression pressure in accordance with the properties of superheated ammonia vapor.

The following tabulation (Table 95) shows how these percentages of volume reduction vary for various initial and final pressures.

In the table of percentage of volume reduction, the pressures are given in lbs. per sq. in. abs. and the corresponding percentages are given as decimal fractions.

TABLE 95.—VOLUME REDUCTIONS DUE TO ADIABATIC COMPRESSION OF AMMONIA.

Abs. Suction Press.	Adiabatic Compression Pressures, Lb. Absolute 25	35	50	75	125	200	300
15	0.324	0.480	0.606	0.715	0.808	....	....
16	0.291	0.453	0.585	0.703	0.798	....	....
17	0.255	0.426	0.565	0.684	0.791	0.855	....
18	....	0.401	0.546	0.671	0.778	0.850	....
19	....	0.375	0.527	0.667	0.770	0.843	....
20	....	0.351	0.506	0.641	0.760	0.834	....
21	....	0.324	0.487	0.626	0.750	0.829	....
22	....	0.302	0.469	0.613	0.743	0.821	....
23	....	0.277	0.451	0.601	0.734	0.816	....
24	....	0.252	0.433	0.586	0.723	0.810	....
25	....	0.229	0.413	0.573	0.715	0.800	....
26	....	0.205	0.396	0.560	0.705	0.797	....
27	....	0.183	0.382	0.540	0.697	0.790	....
28	....	0.157	0.360	0.532	0.688	0.788	....
29	....	0.133	0.342	0.519	0.678	0.780	....
30	....	....	0.328	0.507	0.671	0.775	0.836
31	....	....	0.308	0.485	0.662	0.769	0.831
32	....	....	0.292	0.482	0.654	0.761	0.828
33	....	....	0.275	0.469	0.645	0.757	0.824
34	....	....	0.257	0.457	0.638	0.750	0.819
35	....	....	0.238	0.448	0.629	0.743	0.813
36	....	....	0.224	0.433	0.619	0.738	0.810
37	....	....	0.207	0.421	0.610	0.734	0.806
38	....	....	0.190	0.408	0.603	0.726	0.802
39	....	....	0.175	0.397	0.596	0.721	0.797
40	....	....	0.156	0.385	0.587	0.716	0.794
42	....	....	0.127	0.362	0.572	0.704	0.787
44	....	....	0.094	0.336	0.555	0.693	0.778
50	....	....	....	0.269	0.510	0.660	0.753

Analysis of Indicator Cards.—Since the indicator card is really a pressure-volume diagram it can be readily used to ascertain the operation of the valves, piston rings, etc. If the compression took place in a cylinder which was composed of a non-conducting envelope so that the refrigerant could neither absorb nor reject heat during compression, the compression curve on the indicator card would follow the adiabatic curve. Practically there is some transfer of heat between the refrigerant and the compressor cylinder walls and there are some other factors that affect the location of the actual compression curves. In general, the actual compression curve on the compressor when it is in perfect mechanical condition will be only slightly below the adiabatic curve. The theoretical isothermal compression curve may be used also to some extent as a reference curve.

In general, the actual compression curve of the indicator card will lie between the isothermal and adiabatic curve, being nearest to the adiabatic curve. The figures *ABCD*, *AFCD*, *AECD* of Fig. 204 illustrate indicator cards which have been taken from a slow speed hori-

zontal double acting machine in which the clearance has been reduced to a minimum. The isothermal curve *AH* and adiabatic curve *AG* may be laid out on this card by dividing the length of the stroke represented by the card into ten equal parts, through which ordinates are drawn. The line *IJ* on the foregoing diagram represents the atmospheric pressure line. The absolute vacuum line *OX* is laid out by locating this line at a distance below *IJ* equal to the atmospheric pressure at the time that the indicator card was taken. Of course, these distances must be laid off with the same scale that is used for the indicator card itself.

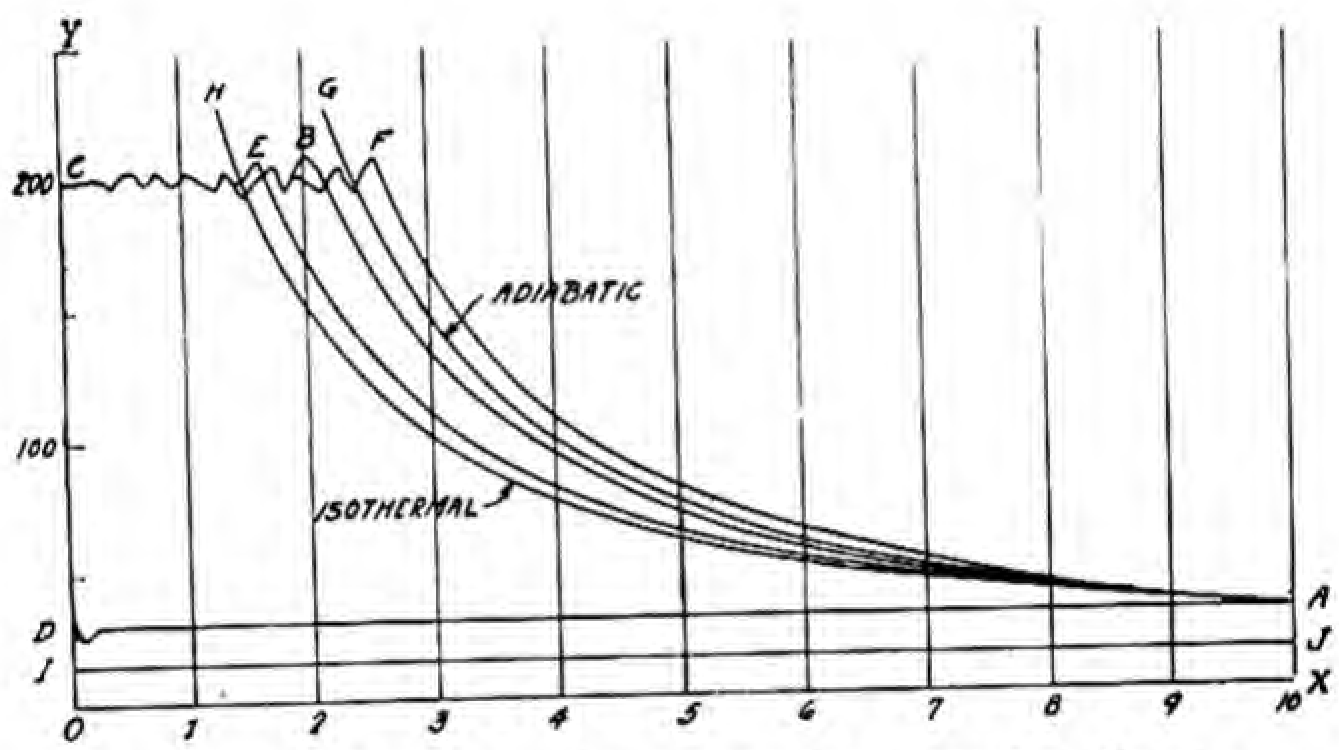

Fig. 204.—Indicator Card with Compression Curves.

By referring to the above, the corresponding adiabatic and isothermal pressures may be selected and laid off on the various ordinates. In this case the compressor was operating between the pressures of 30 and 200 lbs. absolute. By referring to the previous calculations the adiabatic and isothermal pressures corresponding to V_9 are seen to be 34.5 and 33.3. These are laid off on ordinate 9 of Fig. 204, after which the values corresponding to the volume V_8, 40.2 and 37.5 are laid off on ordinate 8, etc., until the theoretical adiabatic and isothermal compression curve *AG* and *AH* are located. After the various points on these ordinates have been located, a smooth curve may be drawn through them. The general relation between the adiabatic curve *AG* and the actual compression curve when the compressor is in first-class mechanical condition is shown by the location of the actual compression curve *AB*.

It will be noted that the actual compression curve is only slightly below the theoretical adiabatic compression curve. An indicator card

such as *AFCD* may be obtained under some conditions of operation. By inspection of the compression curve *AF* in comparison with the adiabatic curve *AG*, it will be noted that the pressures build up faster than they should as the piston proceeds through the stroke. This could be due to the fact that the discharge valve is leaking, allowing the high pressure gas to come back into the cylinder, which will cause the pressure to rise above the normal. Nearly the same effect could be caused by leakage of the gas past the piston rings. Sometimes a card similar to *AECD* is obtained. By comparing the actual compression curve *AE* with the adiabatic curve, it will be noted that the pressures are not as great as they should be as the piston progresses through the stroke. This would indicate that the gas is leaking out of the cylinder in some manner. The leakage is generally through the suction valve on the low pressure side of the system.

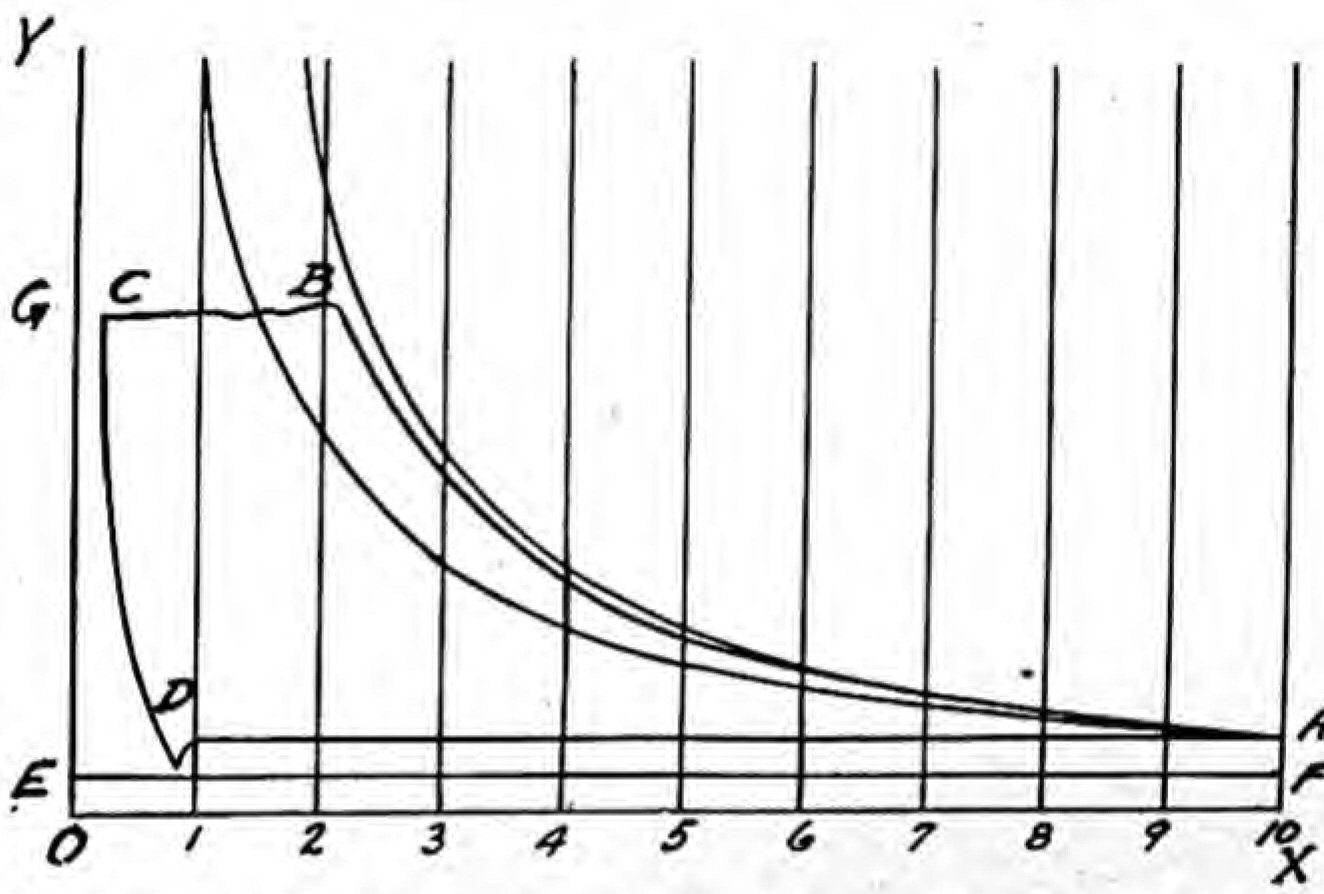

Fig. 205.—Indicator Card with Curves and Showing the Effect of Clearance.

The foregoing method of analysis may be applied to indicator cards which have been taken from compressors that have clearance. These are generally the high speed compressors. The diagram *ABCD* of Fig. 205 is an indicator card of this type. The absolute pressure or vacuum line *OX* is laid off approximately 15 lbs. below the atmospheric line *EF*. The clearance line *OY* is laid off by making the distance *CG* equivalent to the clearance in the compressor cylinder. After the clearance line *OY* and the vacuum line *OX* have been established the vacuum line *OX* is divided into ten equal parts as indicated by Fig. 205. After this has been done, the theoretical adiabatic and isothermal pressures corresponding to the various volumes may be calculated as previously indicated at a point corresponding to the suction pressure.

It is well to note what defects can be discovered by observing the line on the diagram representing the discharge of the gas from the cylinder. If the discharge valve sticks or binds, it is evident that the pressure in the cylinder must be increased to a considerable extent above that in the condenser in order to open the discharge valve. In Fig. 206 the condenser pressure is shown by line *AB*. The distance

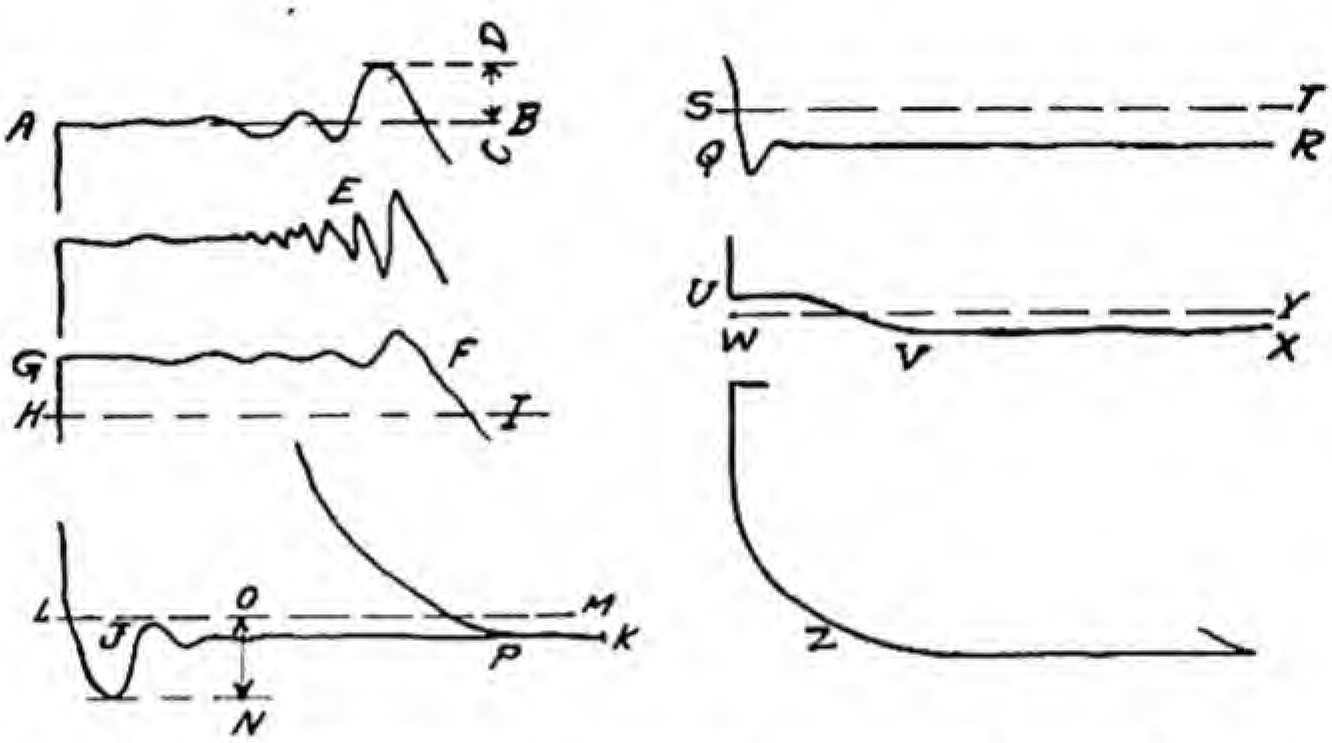

Fig. 206.—Compressor Defects Shown on Indicator Cards.

CD indicates how much the pressure must be increased above the condenser pressure in order to open the valve when it is binding or sticking. The tendency of the discharge valve to vibrate is shown at *E*. This same form of curve may be also sometimes due to the defects of the indicator mechanism. In some compressors the gas may encounter too much resistance as it is being forced out of the cylinder. The velocities of the gas through the passages may be exceedingly high, the spring on the valves may be too strong, and the valves may be quite heavy. Under these conditions the pressure during the discharge of the ammonia from the cylinder will be considerably above the condenser pressure. A condition of this sort is shown along line *FG*. Line *HI* represents the condenser pressure, while line *FG* represents the average discharge pressure in the cylinder.

In a similar manner, the indicator diagram can be used to discover defects of operation in the suction valves. Line *JK* shows how the pressures will vary when the suction valves stick or bind, so that the pressure must be reduced considerably below that in the suction connections in order to open the valve. Line *LM* represents the pressure in the suction main, and the distance *ON* represents the depression of the pressure required to open the valve. If the suction valve sticks or binds it may not close until the piston has started on the compres-

sion stroke for a considerable distance. This is shown at point P, at which point the diagram shows that the suction valve has closed and that the compression of the gas has begun.

In a manner similar to the discharge valve, the suction valve may be constructed so that the areas are limited, the springs may be too strong, and the valves may be heavy. This means that the pressure in the cylinder must be considerably below the pressure in the suction line in order to force the gas into the cylinder against the resistances. This relation is shown by line QR. Line ST represents the pressure in the suction main and line QR represents the pressure in the cylinder during the suction stroke. Sometimes, if there is a considerable leakage of the gas past the piston, the pressure in the cylinder will not be reduced enough to allow the suction valve to open until the piston has traversed a considerable portion of the stroke. This condition is shown by line UVX, while line WY represents the pressure in the suction main. Leakage of high-pressure gas past the piston or discharge valve will give the rounded expansion line Z.

In addition to the foregoing, the operating engineer is able to discover many other minor defects by means of the indicator.

Working Temperatures and Pressures in the Condenser.—One of the most important steps in the building and operating of an efficient refrigerating plant is that concerning an ammonia condenser which is adapted to the size and type of refrigeration system. The selection of the type of condenser to be used and the amount of condenser surface to be required depend upon several factors, of which the following are probably the most important: (1) Temperature of water supply; (2) amount of water available; (3) nature and source of water supply; (4) location of condenser and plant; (5) capacity of plant in tons of refrigeration; (6) cost of pumping condenser water.

Removal of Heat by the Condenser.—Before observing the effect of these factors on the size and type of condenser to be used, it might be advantageous to investigate the heat-transmission phenomena which underlie the operation of the condenser. The fundamental physical law underlying the operation of the condenser states that heat always tends to flow from a body of higher temperature into a body of lower temperature. The amount of heat removed by a condenser of certain dimensions under given conditions depends on the area of the effective condenser surface, the rate of heat transfer through a square foot of condenser surface in a given time for one degree of mean temperature difference of the ammonia and water, and the mean temperature difference between the ammonia and water. Then, with a given amount of surface (having a certain heat-transfer rate), it is

necessary to have a mean temperature difference large enough to produce a flow of heat that will condense the ammonia. The amount of heat removed by the condenser may be expressed as follows:

$$H = K \times A \times M.T.D.$$

in which H = heat removed by the condenser, expressed in Btu. per hour;
A = area of the effective condenser surface;
K = heat transfer coefficient, which is the rate of flow of heat in Btu. per sq. ft. per degree temperature difference per hour:
M.T.D. = mean temperature difference between the ammonia and the water.

In the superheated portion of the condenser, the heat flows from a gas to a liquid; in the liquefying part, from a condensing vapor to a liquid; and in the aftercooling portion, from a liquid to a liquid. On account of these conditions, the heat-transfer coefficient for these portions of a condenser have different numerical magnitudes, so the heat-transfer formula must be applied to each portion separately.

Temperatures in the Condenser.—By noting the distribution of heat in the various parts of the condenser and by observing the mean temperature differences in these parts, the effect of the amount of water available and its temperature upon the selection of a condenser will more readily be understood.

As an example, an ammonia compression system working between 5° (33.79 lbs. abs.) and 86° (170.2 lbs. abs.) may be taken, when the liquid ammonia is aftercooled to 76° F. and a double pipe ammonia condenser is used. If the condenser water has an initial temperature of 70° F. and is heated to 82° in passing through the condenser, the following data* may be developed:

TABLE 96.—CONDENSER DATA.

1. Temperature at end of compression, degrees F.	213.5
2. Temperature in saturated portion of condenser, degrees F.	86.0
3. Temperature of aftercooled liquid, degrees F.	76.0
4. Heat of superheat, B.t.u.	82.4
5. Heat of liquefaction, B.t.u.	497.5
6. Heat of aftercooling, B.t.u.	11.6
7. Heat removed in condenser, B.t.u. per lb.	591.5
8. Heat removed by water, B.t.u. per lb.	12.0
9. Lbs. of water per lb. of ammonia	49.3
10. Temperature of water at inlet of aftercooler portion	70.0
11. Temperature of water at outlet of aftercooler portion	70.24
12. Temperature of water at inlet of superheater portion	80.32
13. Temperature of water at outlet of superheater portion	82.00
14. Temperature difference at inlet of aftercooler portion	6.00
15. Temperature difference at outlet of aftercooler portion	15.76
16. Temperature difference at inlet of superheater portion	5.68
17. Temperature difference at outlet of superheater portion	131.5
18. Mean temperature difference aftercooler, $m.t.d._a$	10.1
19. Mean temperature difference liquefier, $m.t.d._l$	9.85
20. Mean temperature difference superheater, $m.t.d._s$	40.00

* Based on Goodenough-Mosher tables.

This information is presented graphically by Fig. 207, which is a heat-temperature diagram for a double-pipe ammonia condenser operating under the foregoing conditions of pressure and temperature. The line *ABCD* represents the change of temperature and heat content as the ammonia passes through the condenser. Thus in the superheater portion of the condenser the ammonia is cooled from 213.5° to 86° by the removal of 82.4 Btu.; in the liquefier portion in changing from vapor to liquid at 86°, 497.5 Btu. are extracted; and in the aftercooler

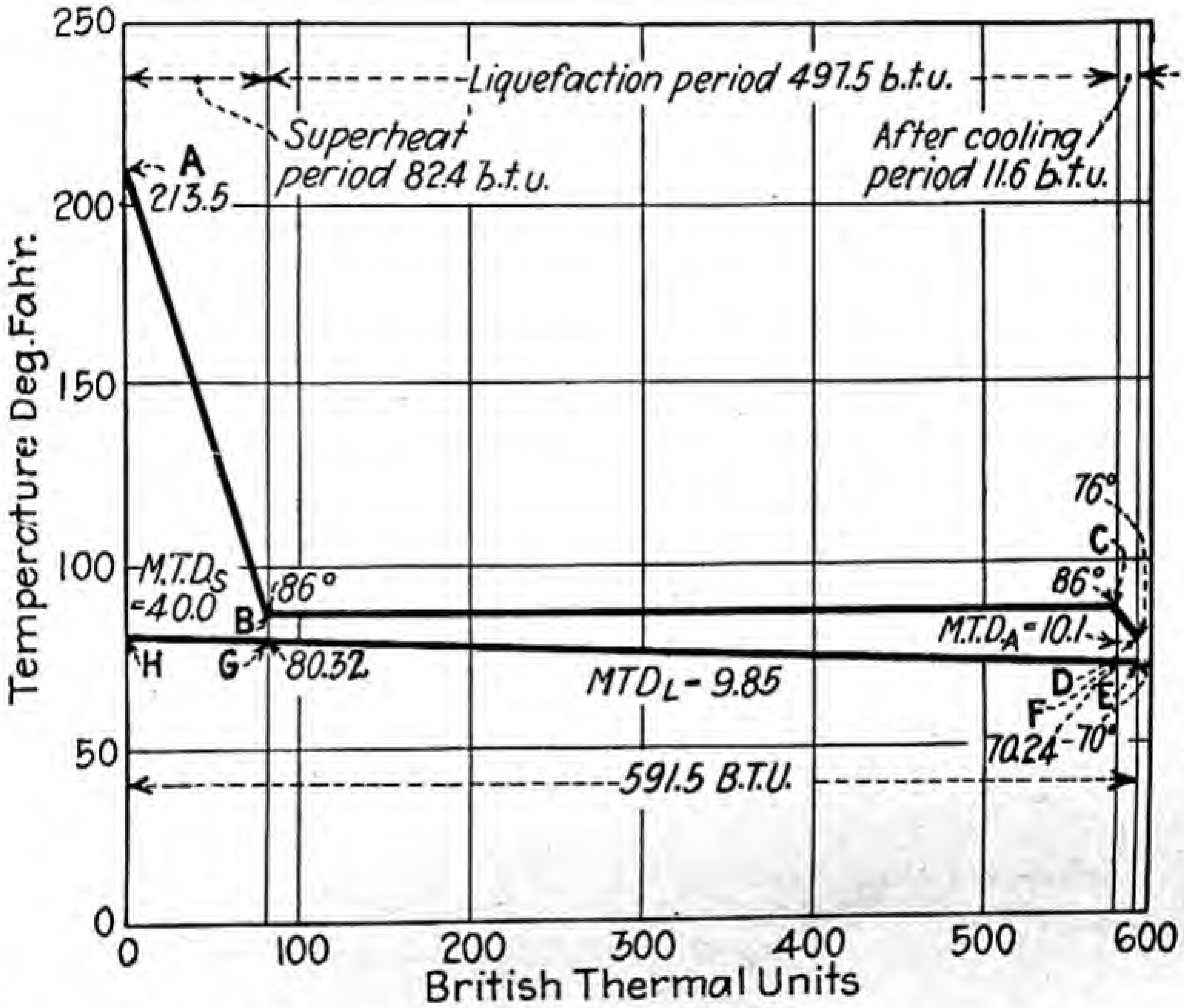

Fig. 207.—Heat Temperature Diagram for Double-Pipe Ammonia Condenser.

portion the ammonia liquid is cooled from 86° to 76° by the removal of 11.6 Btu. The line *EFGH* represents the change of temperature and the heat removed by the condenser water. The water rises from 70° to 70.24° in passing through the aftercooler portion, 72° to 80.23° in the liquefier, and 80.32° to 82° in the superheater portion.

Temperature of Condenser Water.—With the ordinary amount of condenser water that is available and with a condenser of the usual amount of surface, the temperature of the ammonia must be high enough above the temperature of the water to produce the flow of heat across the condenser surface. This may be appreciated by an

inspection of Fig. 207 and the fundamental heat transfer formula. The liquefaction portion may be examined as an example. Line *BC* represents the ammonia and line *FG* the water temperature. The mean temperature difference is equal to 9.85°. Then, by using the heat-transfer formula, the product of this mean temperature difference, M.T.D., the heat transfer coefficient, K, and the area of the surface, A, will give the total quantity of heat, H, removed by the condenser surface.

Using water at temperatures higher than 70° would elevate lines *BC* and *FG*, and water of lower temperature would depress these lines. The elevation of line *BC* means higher condenser pressures. Generally speaking, it may be said that the mean temperature differences are diminished in actual practice as the water temperature increases. This means that the condenser using water at a high temperature will have more surface per ton of refrigeration capacity than one using cool water. Under average conditions the temperature of the water leaving the condenser will be from 5° to 10° below the temperature of the ammonia in the liquefying portion of the condenser. Generally speaking, it may be said that atmospheric and vertical shell-and-tube condensers are used when only very warm condenser water is available, while the double-pipe and closed shell-and-tube types may be used when cooler water is available.

Amount of Condenser Water Available.—With an abundant supply of cooling water the temperature range may be reduced to the minimum, which tends to reduce the condenser pressure. Reducing the temperature's range places the line *EH* in Fig. 207 in a more nearly horizontal position. Likewise with the water scarce, the temperature range is greater and the line *EH* rises more sharply. It is apparent that in this case the condenser pressure will be higher.

When the water supply is abundant, any type of condenser suited to the plant may be used, but when the water is scarce it may be found advantageous to select the double-pipe or closed shell-and-tube type using a long range of temperature on the water.

The great benefit of evaporation with the atmospheric condenser is that it reduces the range of the water temperature. This in turn reduces the temperature of the condensing ammonia, which results in reduced condenser pressure.

It must be remembered that there are three important factors that affect the condenser pressure which may be obtained in any plant; namely, the amount of effective condenser surface, the temperature of the water supply and the amount of water available for condensing purposes. The installation of a large amount of condenser surface is to be advocated for almost every plant. The installation of an addi-

tional amount of condenser surface is a question of initial cost, and the saving that is obtained by a reduced condenser pressure is, of course, a continual one.

The water for the condensers is taken usually from one of the following sources: (1) Deep wells; (2) rivers, lakes, or canals; (3) cooling ponds or towers; (4) the sea.

The waters that are pumped out of wells in the earth vary greatly in general makeup, which is especially true of the mineral matter content. Water containing a comparatively large amount of the dissolved minerals is commonly termed "hard." Now, these hard waters, upon being heated as they pass through or over a condenser, may deposit scale or sediment on the condensing surfaces.

The method of pumping also has a bearing on the precipitation of mineral matter from the water. A large amount of solids may be deposited by hard waters that are pumped by means of the air-lift pump. This is especially true if an air separating tank is not installed.

From the foregoing, the conclusion may be drawn that if scale is deposited by hard water below 100° F., it is necessary to use the atmospheric types of condenser. The scale may be easily removed from condensers of this type. If the water is not very hard and deposits only a soft sediment or film of matter below 100° F., the double-pipe type condenser may be used.

Water taken from rivers, lakes and canals is ordinarily considered "soft"; that is, it contains a comparatively small amount of dissolved mineral matter. On the other hand, the water generally contains a good amount of mud, leaves, bits of wood, vegetable matter and other suspended material. Thus, in using this kind of water in a condenser, the trouble is not so much from the formation of scale as from the deposit of the suspended matter. Therefore, it is generally desirable to install the atmospheric or the vertical shell-and-tube type of condensers on the plants using this sort of water, in order to facilitate the cleaning process.

Water for the larger-sized refrigerating plants is sometimes recirculated and recooled by pumping it over a cooling tower or through a set of spray nozzles over a spray pond. In this system the water tends to be contaminated with algae, moss and other organic matter. Although either type of condenser may be used with success, preference is generally given to the open shell-and-tube or atmospheric type, since this type is more easily kept clean. Of course, strainers, screens and other separating devices are inserted in the pump suction lines to prevent any large pieces of matter from being discharged to the condenser.

Refrigerating plants located along the seashore may well take their condensing water from the ocean. Sea water generally does not throw

down a heavy precipitate of scale-forming solids, so that either type of condenser may be used with success. With comparatively cool sea water, the double-pipe condenser gives excellent service under these conditions. While not absolutely essential to the operation of the condenser, it is generally advisable to use galvanized pipes to avoid the corrosive action of the sea water on the condensing surfaces.

Location of Plant and Condenser.—The particular locality of the country where the condenser is installed may have a bearing on the selection and operation of the condenser. It has previously been pointed out that the cooling due to evaporation of a part of the condenser water is of considerable importance. This is especially true in localities where the moisture content of the air (relative humidity) is low. Therefore, it is entirely desirable to give the atmospheric type of condenser preference in these localities. In the more humid climates, it is often advantageous to use the closed types when it is possible.

The space and relative location of the condenser coils often have a direct bearing on the type of condenser that should be used, and its operation. When space is available on a roof or the outside of the building, it is feasible to use the atmospheric type for reasons previously indicated. When condensers are installed in this manner, the coils should be arranged so that the air will have free movement between the pipes, and the pipes should be protected from the direct rays of the sun.

Oftentimes atmospheric condensers are installed in connection with cooling towers and spray ponds. The relative position of the water-cooling apparatus and condenser coils should receive especial attention. These should be placed one above the other and preference should be given to placing the condenser on top, if possible, in order to secure free air movement around the coils, since air must be removed from the spaces between the coils as soon as it has absorbed some moisture, for efficient operation. When it is desirable to place the condenser in the machine room, it is preferable to use the shell-and-tube or double-pipe type. This eliminates all splash and dampness in the machine room; thus the condenser may be placed in part of the building.

Economic Considerations.—The matter of the proper selection of an ammonia condenser with its water-circulating apparatus and the efficient operation of same is a consideration of economic importance. The condenser and its auxiliaries should be selected with the idea of securing the greatest economy over a period of years equal to the life of the condenser. In the ultimate analysis it will be observed that it is desirable to reduce to a minimum the cost of a ton of refrigeration.

If an excessive amount of condenser equipment is installed, the initial cost is very large and a larger amount of cooling water must be supplied. Thus the interest on the additional investment, cost of pumping the larger amount of water, etc., increase the cost of operation. Again, if an insufficient amount of condenser equipment is installed, the condensing pressure will be extraordinarily high, which will result in the expenditure of an excessive amount of work to compress the ammonia. This, of course, increases the cost of power to operate the plant.

TABLE 97.—AMMONIA CONDENSER PRESSURES (COOLING TOWER WATER).

	Dry Bulb Temperature	Relative Humidity	Wet Bulb Temperature	t_1	t_2	t_a	Condenser Pressure
Jan.	32.0	74.3	29.6	41.0	51.0	56.0	85.2
Feb.	39.5	73.9	36.4	46.0	56.0	51.0	94.2
Mar.	43.5	71.4	39.7	49.0	59.0	64.0	101.0
Apr.	57.0	65.4	51.8	57.0	67.0	72.0	118.7
May	66.0	68.2	59.0	63.0	73.0	78.0	133.2
June	75.5	68.2	66.8	71.0	81.0	86.0	154.5
July	79.5	66.1	70.7	74.0	84.0	89.0	163.0
Aug.	77.5	67.5	69.5	73.0	83.0	88.0	160.1
Sept.	70.0	68.6	62.7	67.0	77.0	82.0	143.6
Oct.	59.0	66.2	52.8	58.0	68.0	73.0	121.0
Nov.	49.5	70.2	45.0	54.0	64.0	69.0	111.8
Dec.	36.5	73.6	33.5	45.0	55.0	60.0	92.9

D.B., R.H. W.B., for St. Louis, Mo.
$t_2-t_1=10°$
$t_a-t_2=5°$
t_1-W.B.$=4°$ to $11°$
t_1=water to condenser
t_2=water from condenser
t_a=ammonia condensing temp.

Ammonia Condenser Pressures.—The variation of ammonia condenser pressures for various months of the year is shown in Table 97. The data is based on the use of cooling tower water, with a 10° F. increase of temperature of the water as it passes through the condenser. It is also assumed that the condenser water will be cooled within 4° F. to 11° F. of the wet bulb temperature.

Frequently the condensing pressure is higher than it should be because non-condensable gases are mixed with the refrigerant gas. Although it was formerly believed that non-condensables originated principally from the breakdown or dissociation of the refrigerant, it is now known that the greatest proportion is air which has been drawn into the system through glands or seals, while repairs are made or in other fashion.

The foreign gas collects on the high side in the condenser and receiver and adds its pressure to that of the refrigerant so that the compressor must work against a higher pressure and this consumes

more power. Non-condensable gases can be purged directly from the system but the process wastes refrigerant and is not completely effective. For this reason, wherever the annual power bill runs up to 500 dollars or more it is usually advisable to install a refrigerated non-condensable gas purger.

This apparatus is really a stripping condenser into which the mixed non-condensables and refrigerant gas are drawn and cooled considerably below the saturation temperature of the refrigerant. The refrigerant is condensed to a liquid while the non-condensables collect above and are discharged to the atmosphere. As the liquid refrigerant collects in the purger it is returned into the system. Modern purgers are automatic in operation and very effective in removing non-condensable gas from the system.

The purger usually is connected to both the condenser and the receiver so that the foreign gas mixture can be drawn from either one. The best operating results are obtained by purging from the receiver if the liquid drain from the condenser is large enough to provide space for gas passage above the stream of liquid and there are no traps in the liquid line.

Frequently a check of the liquid temperature and condensing pressure in a plant will show that non-condensables are causing excess head pressure of 20 lbs. or more. A general approximation is that each 2 lbs. of excess pressure waste 1 per cent of the power consumed by the compressor. That is an important item in plant costs. The presence of excess pressure is determined by checking the temperature of the refrigerant flowing out of the condenser with the saturation temperature corresponding to the existing condensing pressure.

In an ammonia system for example the temperature of the liquid leaving the condenser is 84° F. and the head pressure is 167 lbs. gauge. According to the ammonia tables the saturation pressure at 84° F. is 149 lb. gauge so that the excess pressure indicated above is 167-149 or 18 lbs. While it is usually not practical to eliminate all of this excess pressure a refrigerated purger would reduce it to about 4 to 6 lbs. and cut the condensing pressure to about 154 lbs. The savings in power would approximate $(167 - 154) \div 2$ or $6\frac{1}{2}$ per cent.

QUESTIONS ON CHAPTER XX.

1. In refrigerating systems, which is the most important, pressure or temperature? State the reasons for your answer.

2. What three factors determine the amount of heat which is transferred in any apparatus?

3. In what way and in what magnitude should the temperature difference between the room, and the ammonia or brine vary?

4. How is the suction temperature in the refrigerating plants controlled and measured?

5. How are the discharge temperatures controlled and measured?

6. How may the pressures existing at different parts of the compressor cylinder be determined?

7. What is the relation between the pressure and volume of ammonia in compressor cylinder?

8. How are the isothermal and adiabatic curves constructed and used to study the conditions inside of the compressor cylinder?

9. Explain fully how the temperatures of ammonia and water vary in the condenser.

10. What factors affect the condenser pressure, and in what manner?

CHAPTER XXI.

PRIME MOVERS FOR REFRIGERATING PLANTS*.

General Considerations.—The proper selection of suitable prime movers for ice making and refrigerating plants is a proposition of economic importance. This is due to the fact that the cost of power is an important item in the total cost of producing ice or refrigeration. It is desirable to produce the maximum amounts of ice and refrigeration with the minimum expenditure of money. Thus the selection of a suitable prime mover and the relative cost of power must receive careful consideration. The ultimate success of a given plant may depend upon the judgment used is selecting the driving power.

In the earlier days of the ice making and refrigerating industries, steam engines were used almost to the exclusion of all other forms of power. This condition was due primarily to the fact that the steam engine was the only practical prime mover available during those times. But, as the industry grew with time, other types of prime movers were introduced. These were the electric motors, gas and oil engines. The use of electric motors and internal combustion engines as prime movers for refrigerating plants has been due to two factors: First, the perfection and inherent efficiencies of these newer forms of prime movers; second, the introduction of raw water ice manufacture. At present, the electric motor is being used extensively for driving this type of industrial plant.

In the ultimate analysis, it will be noted that economy of operation is one of the most important factors. In some cases, it is desirable to install steam engines. The steam engine has been improved, also, as the industry grew. At present, efficient engines of the poppet four valve and uniflow type are available. The uniflow engine especially is receiving attention, due to the fact it uses steam economically over a wide range of loads.

On the other hand, in many cases electric motors may be used advantageously. Motors having high efficiencies may be obtained in

* Considerable matter in this chapter has appeared in articles written by the author for *Power* at various times, and permission has been granted for its re-use here.

all parts of the United States. The synchronous motor is being used extensively for the larger installations, on account of its ability for direct connection to compressors and its high operating efficiency.

Therefore, the proposition of the suitable selection and economical operation of prime movers for refrigerating plants must receive careful consideration from all possible viewpoints. It is evident that each particular plant must be considered individually in order to arrive at the most successful solution.

Steam Engines.—The proper selection and the efficient operation of steam engines for driving refrigerating plants are therefore considerations that have great economic importance. It is desired to produce a ton of refrigeration with the least expenditure of money and time. The cost of maintaining the engine room in operation, the cost of fuel that is used for power purposes, and the depreciation on the mechanical equipment are important items that enter into the determination of the final cost of a ton of refrigeration.

The most important practical considerations entering into the selection of the prime mover are the folowing: (1) Reliability; (2) economy of operation; (3) simplicity; (4) flexibility of operation; (5) accessibility; (6) depreciation; (7) space requirements; (8) character of refrigeration plant.

It is very apparent that the refrigeration plant is a type that must have a dependable and reliable method of driving. The large cold storage warehouse may be considered as an example. These warehouses are filled with perishable food products generally. It is easy to imagine the great loss that would occur should the refrigeration service fail, due to a breakdown of the prime mover that drives the refrigerating machine.

Factors That Determine Engine Type.—The principal factors that affect the economy of the operation of the prime mover are the cost of the fuel, the relative amount that is consumed, and the cost of maintaining the engines in operation. The cost of the fuel should be kept as low as possible, but on the other hand, it is generally advisable to secure as good a fuel as is ordinarily obtainable. The fuel consumption should be as nearly constant as possible under varying load conditions. This should be true especially between one-half and full-load capacities.

Ease of operation is an important consideration under all circumstances. The engines should be easy to start and stop. They should be simple in construction so as to be readily understood by the average operating man. If the various parts of the engines are accessible, they are subjected to more thorough inspection, which tends to eliminate breakdowns, and in the case of a mishap the repairing of the

damaged parts is facilitated. The parts themselves should be simple in construction, which promotes ease of demounting and reassembling.

The life of any apparatus is an economic consideration. Preference should be given to equipment having the slowest rate of depreciation in order that the disadvantages of deterioration and consequent loss of efficiency may not become too great.

Local Conditions Oftentimes Determine Type of Prime Mover.—The particular type of prime mover to be employed is oftentimes determined by the local conditions existing in the plant itself. There may be near the plant a cheap supply of fuel.

The type of steam engine that has been most generally used is the simple non-condensing Corliss, although it has been largely superseded by the compounded Corliss. The wide use of the steam engine has been due to several factors, among which the following two are the most important: In the first place, it was not until very recent years that a cheap and dependable power in the form of electricity was available for refrigerating plant service. Thus, in the past it has been necessary to develop the power for driving the refrigerating machine directly in the plant itself, which condition leads to the use of the steam engine. In the second place, ice making plants were required to use distilled water in order to produce clear and merchantable ice. To obtain these large quantities of distilled water easily and economically, the exhaust steam from the main driving engines was condensed. This necessitated the use of a steam engine. Of course, this condition existed only before the development of the raw water ice making system. With this system any type of prime mover may be employed, so that at present, electric motors are being used instead of the steam engine as a method of driving the plants in some installations. However, even at present there are many conditions which warrant the installation of steam engines. The type of prime mover that is best suited to a given refrigeration plant is the one that delivers the power at the least cost, taking into consideration all the various charges, such as fuel, labor, supplies, repairs, interest on investment, depreciation, taxes.

Thus, in a locality where the cost of electrical energy is not extremely low, it is advisable to consider the steam engine as a probable prime mover. This is especially true of large plants. If the installation is located near a supply of coal, the steam plant will generally be found to be the economical choice. Or again, steam under considerable pressure may be required in other parts of the plant, which would make the use of steam engines more logical than other forms of prime movers. The kind and cost of the water that is available for power purposes have an important bearing upon the subject. Also, the

relative capacity of the individual plant is an important consideration. It must be observed that each individual installation is a problem in itself and therefore requires its own special prescription.

Steam engines that are used for driving refrigerating plants may be classified as slide-valve, Corliss-valve, poppet-valve and uniflow. They may be further divided upon a rotative-speed basis into high-speed and low-speed. They may be classified as non-condensing or condensing, and may be further separated on the steam-expansion basis into single and compound-expansion types.

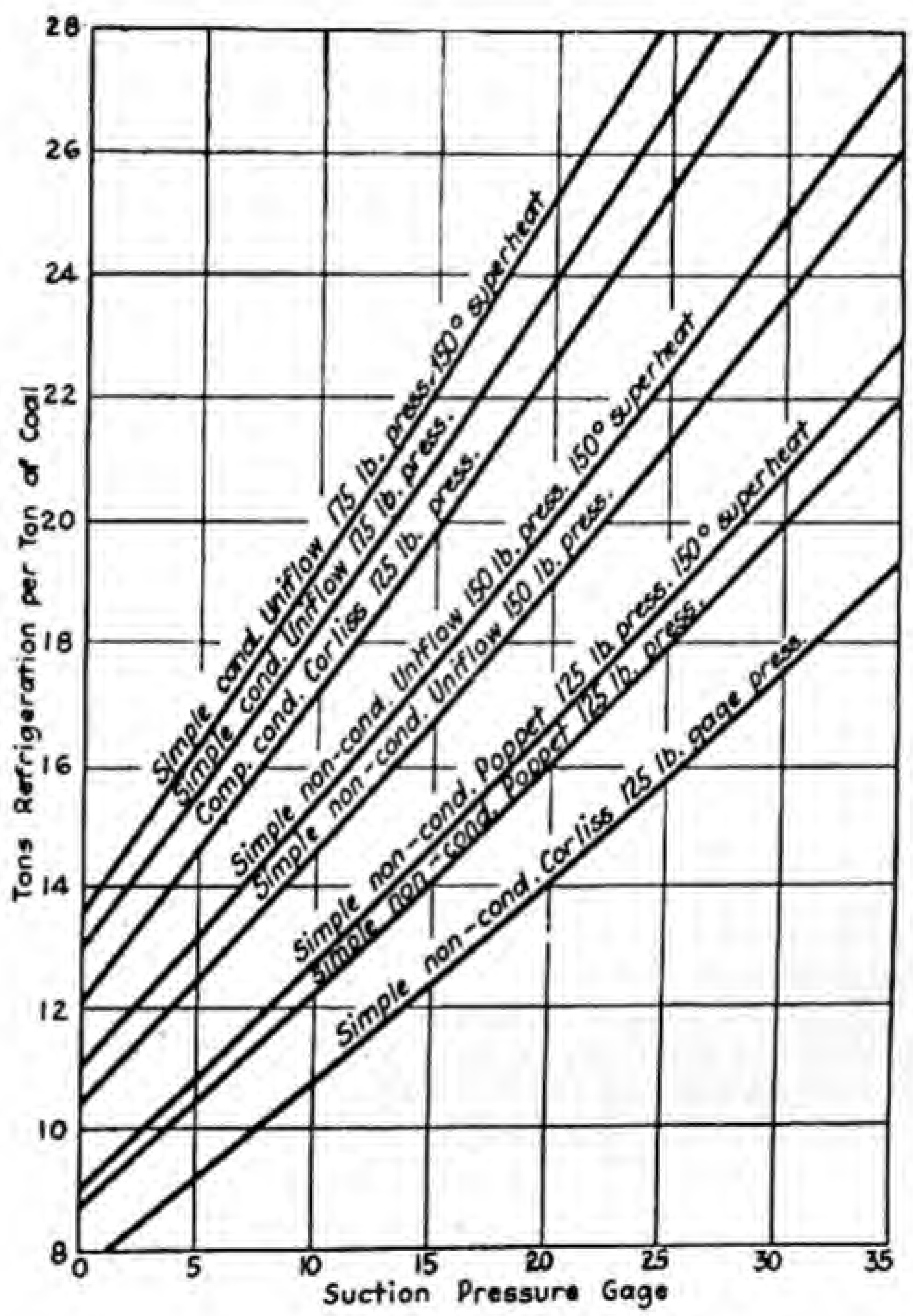

Fig. 208.—Comparative Efficiencies of Plants.

Comparative Efficiencies of Plants.—In order to show graphically the relative efficiencies of the various methods of driving refrigeration plants, Fig. 208 has been prepared. This shows the number of tons of refrigeration that may be produced by the expenditure of a ton of fuel having a heating value of 12,500 Btu. per lb., for various suction pressures. An average condensing pressure of 170 lbs. per sq. in. gauge was assumed. The boiler-plant efficiency was taken as 60 per

cent, and an allowance was made for the usual complement of auxiliaries.

As most plants are operated with Corliss engines, it is well to note the relative efficiency of the simple non-condensing and the compound condensing types. The compound-condensing Corliss-engine plant is very economical in the consumption of steam. On the other hand, a single-cylinder non-condensing uniflow-engine plant with high steam pressures and temperatures will have practically the same efficiency. The equipment for the uniflow plant is much simpler and requires, of course, less auxiliary power. A condensing uniflow-engine plant using superheated steam has a higher efficiency than the complex compound-condensing Corliss-engine-driven plant. The use of uniflow and poppet-valve engines with high steam pressure and temperatures is to be advocated for all sizes of plants.

The popularity of the steam engine as a prime mover of refrigeration plants is due, no doubt, to the fact that it is a reliable and dependable means of producing power. It continues to operate under very adverse conditions and will stand a maximum amount of abuse. The compound condensing engine has the disadvantage of being more complex and, therefore, is more difficult for the average operator to handle efficiently.

As compared to plants which are driven by internal-combustion engines or electric motors, the steam-driven plant requires much more space. Compound engines require a maximum amount of space, while simple single-cylinder engines require a medium amount of space. Space requirements demand important consideration where the cost of land is excessive.

Steam engines have a slower rate of depreciation than the internal combustion engine and probably will have a useful life somewhat longer than electric motors. The useful life of steam engines will vary from 15 to 25 years, depending, of course, upon the keeping of the equipment in good physical condition by proper maintenance and repairs. The hours of operation have only a slight bearing on the amount of depreciation of the engines. If the engines are kept in proper repair, continuous operation will not cause much greater wear and tear than that produced by intermittent operation. This is due to the fact that engines operating intermittently are subjected to temperature stresses, which will produce more depreciation than the mechanical wear that is occasioned by continuous full-load service under careful operation.

Electric Drive.—In the selection of a suitable method for driving refrigeration plants, the local conditions must be given thorough consideration. In many localities the cost of fuel for power purposes is

high or the delivery unreliable, and electric motors are favored on account of lower power cost per ton of refrigeration. Another economic consideration favoring the installation of electric motors is that this method of driving has a comparatively high efficiency and the efficiency remains high at decreased loads. The first cost of the electrically-driven plant is frequently lower than that of a steam plant, and this, in many instances, dictates the choice of electric drive. Furthermore, the total cost of power in the electrically-driven plant will probably be less on account of the fact that less labor and a smaller amount of space are required and overhead charges are somewhat lower.

The advent of the raw water ice making system in 1909 gave a great impetus to the application of electricity in refrigerating plants. Since distilled water is not essential, the compressors, as well as the auxiliaries, can be operated with electric motors. Likewise, since the electrically-driven plant does not require to be near a railroad in order to receive its fuel supply, it may be located in any convenient place near the center of the distribution territory. This reduces very materially the delivery expense of the ice because the truck mileage and driver's time are reduced.

An additional consideration in favor of the electric-drive is that the electric motors in refrigerating and ice plants are considered a desirable load for the central station. This is due to the fact that this kind of electrical load can be arranged so as to produce a more desirable and uniform total load on the central station. The usual plant can decrease its power requirements during the central-station peak-load, enabling a low electric rate to be secured.

Reliability of Electric Drive.—The increasing popularity of the electric-drive is in no small measure due to the reliability of the operation, which is based upon two fundamental factors. In the first place, the engine and boiler in the refrigerating plant are more subject to mishap and shutdown than is the compressor. In the second place, the electric motor is very dependable, being one of the most sturdy, practical, and trouble-proof pieces of apparatus in the power plant.

Of course the motors must be installed properly and they must receive attention and proper lubrication. In the case of power transmission by belting, the belt wheels, motor pulleys and belts must be properly designed to eliminate flipping, etc.

The electrically-driven plant requires the minimum amount of attendance, and the operator does not have to be as high grade as is required for the efficient operation of steam units. The electrically-driven plant is virtually automatic as long as there is continuity in the central station service.

Flexibility of Operation.—Flexibility of operation is easily obtained in the electrically-driven refrigerating plant when it has been properly designed. In the first place, electricity readily lends itself to the subdivided power plant. Thus several units of about the same size may be installed in a given plant and the magnitude of the refrigeration load at any hour will determine the number of these units to be operated. In the electrically driven plant it may be feasible to install, in addition to the main units, a small compressor to pump out any part of the system or to furnish the refrigeration in the winter months, thus allowing the main units to be shut down. The flexibility of the operation of the auxiliary apparatus is one that does not ordinarily receive

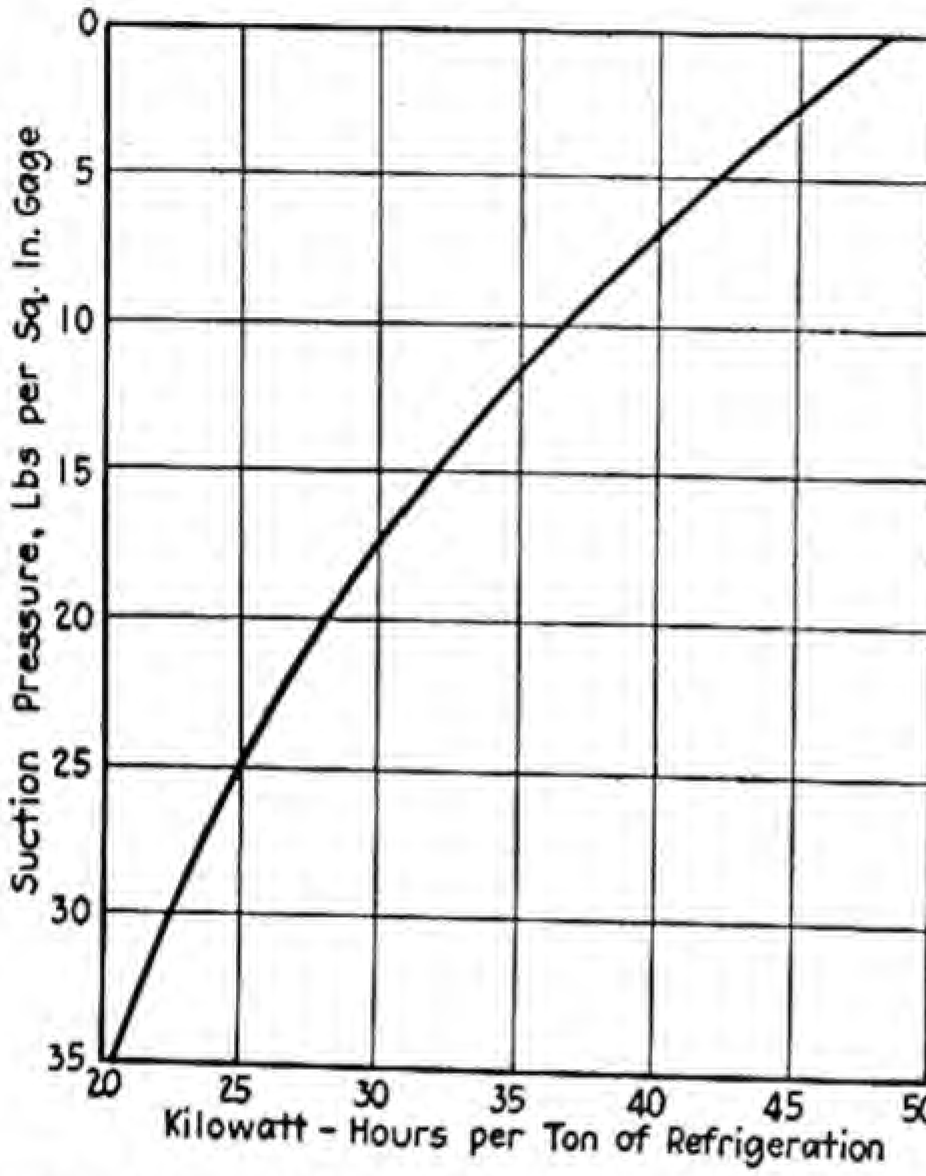

Fig. 209.—Power Required for Ammonia Compressor.

the proper attention. The selection of the auxiliaries with considerations as to their economies for one-half to full-load is important, as well as their practical arrangement in the plant. The correct number of auxiliaries is advocated for each plant. It is obvious that the auxiliaries should be directly connected to their respective motors wherever possible.

Economy of Operation.—The power that is required to drive an ammonia compressor depends upon a number of factors, among which the following are probably the most important: The ammonia work-

ing pressures, the type of compressor and motor, the relative size of motor, and the method of power transmission. Generally speaking, it may be said that the average electrically-driven plant has a comparatively low consumption rate in Btu. per ton of refrigeration. This is probably due to the fact that the actual generation of power in the average refrigeration plant using its own generation machinery is not efficient, while, on the other hand, the central electrical stations have comparatively high efficiencies, and the transmission and utilization of the electrical energy is accomplished in an economical manner. This, however, does not apply to the many steam-driven refrigerating plants equipped with modern condensing engines giving a very high economy. Since the ammonia working pressure determines to a very large degree the amount of power that is required, Fig. 209 has been prepared to show graphically the variation of the power required to operate the compressors at the different suction pressures and at an average condenser pressure of 170 lbs. per sq. in. gauge. Average operating efficiencies were assumed. This chart shows the power that is consumed by the mechanical compression of the vapor alone and does not include any allowance for the usual auxiliaries of the plants. Under the ordinary conditions, the power required to operate the auxiliaries tends to become abnormally high. In the larger plants it is advisable to install an electric meter to measure the current consumed by the compressors, while a separate meter should be used to indicate the current consumption of the motors which drive the auxiliaries.

The effect of the load-factor upon the economy of operation is a very important consideration. The economy of operation decreases as the load-factor is lowered. The load-factor is improved by installing subdivided power units, by providing storage capacity, by operating continuously, etc.

The Selection of a Suitable Motor.—The correct application of electric motors to refrigeration plants depends upon a number of factors, among which the following may be considered important: The operating characteristics of the motors, the type of refrigeration plant, and the capacity of the plant. Among the operating characteristics of the motors, the speed-torque considerations are probably the most important. In general, it will be noted that the local conditions will make each plant a particular problem in itself, so that no specific rules may be given. Therefore, only the general considerations of the types and characteristics of the electric motor will be given.

Generally speaking, it may be said that the alternating-current polyphase induction motors are inherently constant-speed machines. Several methods have been devised to vary the speed of induction motors for a given output, but none of these methods have been entirely suc-

cessful and practical. The speed falls off only slightly as the load is increased, the decrease from no-load to full-load being less than 10 per cent of synchronous speed. The squirrel-cage induction motor has a comparatively small starting torque, while on the other hand the starting current is quite large. The power factor of the starting current is low, and since the starting current is large, heavy fluctuations of the voltages of the supply line are induced. However, this type of motor has the advantages of being simple and strong in construction.

In refrigerating plants this type is used in sizes of 25 hp. and less for driving centrifugal pumps and the like. The starting torque of the slip-ring alternating-current motor is large and the starting current is not excessive. Ordinarily, the starting current for full-load torque is approximately three times the full-load current and the starting torque one and one-half times the full-load torque. Thus the slip-ring motor is particularly well adapted to starting heavy loads with the minimum amount of current. However, heavy overloads should not be carried, since these will produce too much slip between the speed of the motor and the revolving magnetic field, thereby stalling the motor. The starting of these motors is accomplished by means of external resistance. Ampere ratings of a.c. motors is shown in Table 98.

This type is generally used in sizes from 25 to 150 hp. for driving ammonia compressors and other machinery. Smaller sizes are well adapted to drive plunger pumps, cranes, and the like.

Synchronous Motors.—At present there is a tendency toward the extensive use of synchronous motors to drive both slow-speed and high-speed ammonia compressors. Motors of this type are often belted to the slow-speed ammonia compressors, but are more generally directly connected to high-speed compressors. The starting torque is comparatively low, being approximately one-third of the full-load torque. They are not inherently self-starting and are started really as induction motors. On the other hand, when they are at synchronous speed, they can carry heavy overloads for short intervals very easily.

The advantages of using synchronous motors directly connected to ammonia compressors should be obvious. The directly connected unit is simple and compact, occupying the minimum amount of space. The expensive belt is eliminated. The motor requires no base, pulley or bearings, since it is mounted directly on the compressor shaft. The rotor of the motor acts as a flywheel so that the compressor flywheel may be lighter than ordinary.

It is also true that this type of motor has a high efficiency. This is quite noticeable when compared to induction motors at the lower speeds. They operate at a high power factor, and consequently a lower

TABLE 98.—AMPERE RATINGS OF A. C. MOTORS.

H. P.	Single Phase Volts				Two Phase Volts					Three Phase Volts				
	110	220	440	550	110	220	440	550	2200	110	220	440	550	2200
¼	4.	2.	1.	.8	......	......	......	......	......	......	......	......	......	......
½	7.5	3.75	1.9	1.5	3.8	1.9	.95	.8	......	4.4	2.2	1.1	.9	......
¾	10.	5.	2.5	2.	......	......	......	......	......	......	......	......	......	......
1	12.5	6.25	3.2	2.5	7.	3.5	1.75	1.4	......	8.	4.	2.	1.6	......
1½	.18	9.	4.5	3.6	9.	4.5	2.25	1.8	......	10.3	5.1	2.6	2.1	......
2	24.	12.	6.	4.8	11.	5.5	2.75	2.2	......	12.5	6.25	3.2	2.5	......
3	34.	17.	8.5	6.8	15.8	7.9	4.	3.2	......	18.	9.	4.5	3.6	......
4	43.	22.	11.	8.6	21.	10.5	5.3	4.25	......	24.	12.	6.	4.8	......
5	55.	28.	14.	11.	26.2	13.1	6.6	5.25	......	30.	15.	7.5	6.	......
7½	80.	40.	20.	16.	38.6	19.3	9.7	7.75	......	44.	22.	11	8.8	......
10	105.	53.	25.	21.	50.8	25.4	12.7	10.2	......	58.	29.	14.5	12.	......
15	145.	73.	36.	29.	72.	36.	18.	14.5	4.0	82.	41.	20.5	16.5	4.5
20	200.	100.	50.	40.	96.	48.	24.	19.2	5.25	110.	55.	27.5	22.	6.
25	250.	125.	63.	50.	120.	60.	30.	24.	6.6	136.	68.	34.	27.2	7.5
30	295.	148.	74.	59.	140.	70.	35.	28.	7.4	160.	80.	40.	32.	8.5
40	370.	185.	93.	74.	184.	92.	46.	37.	9.6	212.	106.	53.	42.5	11.
50	......	......	......	......	230.	115.	57.5	46.	11.8	264.	132.	66.	53.	13.5
60	......	......	......	......	276.	138.	69.	55.	14.	316.	158.	79.	63.	16.
75	......	......	......	......	344.	172.	86.	69.	17.5	392.	196.	98.	78.5	20.
100	......	......	......	......	454.	227.	114.	91.	22.8	518.	259.	130.	104.	26.
125	......	......	......	......	556.	278.	139.	111.	2.8	636.	318.	159.	127.	32.
150	......	......	......	......	666.	333.	167.	133.	33.	760.	380.	190.	152.	38.
175	......	......	......	......	779.	390.	195.	156.	39.	890.	445.	223.	178.	45.
200	......	......	......	......	892.	446.	223.	178.	45.	1020.	510.	255.	204.	51.
250	......	......	......	......	1086.	543.	272.	217.	55.	1250.	625.	313.	250.	62.5
300	......	......	......	......	1314.	657.	329.	263.	66.	1500.	750.	375.	300.	75.
400	......	......	......	......	1750.	875.	438.	350.	87.5	2000.	1000.	500.	400.	100.
500	......	......	......	......	2180.	1090.	545.	436.	109.	2500.	1250.	625.	500.	125.

electrical rate is usually given plants having synchronous motors.

This type should generally be used for driving compressors when the horsepower requirements are above 50. The present tendency is to use synchronous motors directly connected to high-speed ammonia compressors. Its one drawback which is shared by all alternating current motors, is the impossibility of securing speed variation; the compressor must be operated at constant speed at all loads.

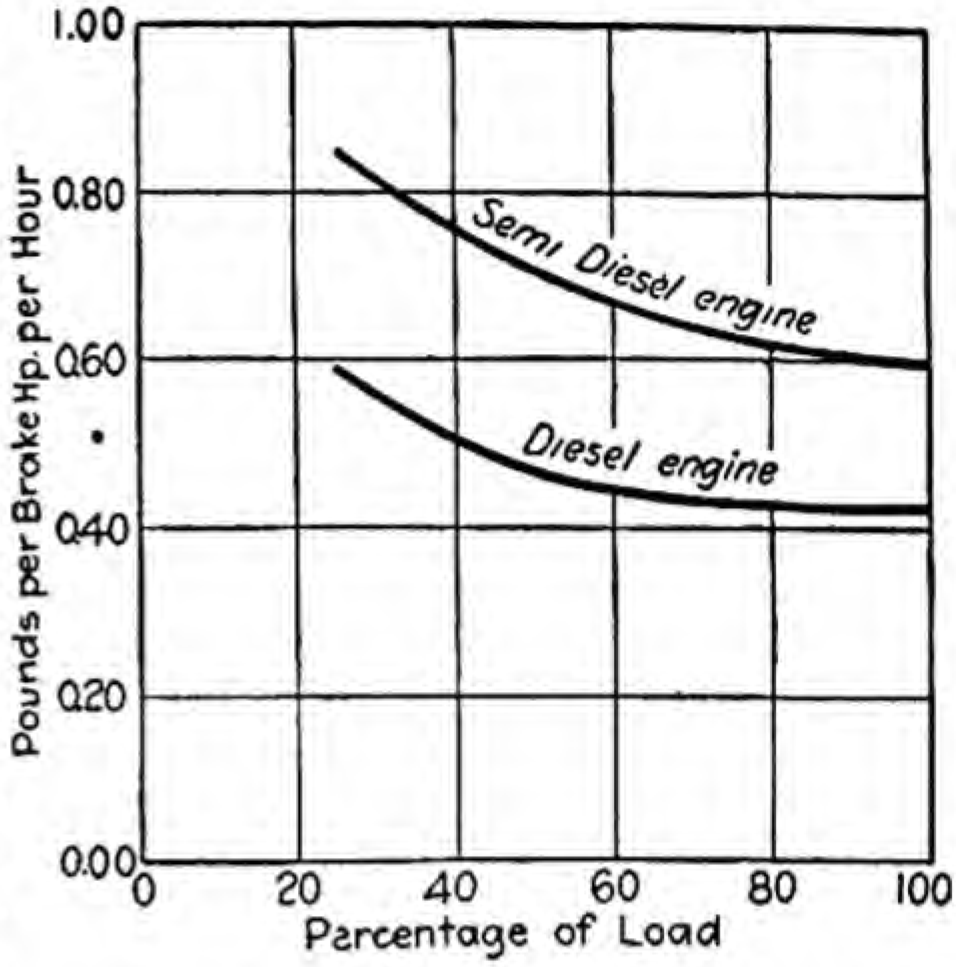

Fig. 210.—Oil Consumption of Oil Engines.

Oil Engines for Refrigerating Plants.—Oil engines have come into wide use for driving refrigerating plants. The type of the plant and its location will have an important bearing on the selection of the most suitable prime mover. It happens sometimes that the refrigeration plant and its auxiliary equipment are placed in such close quarters that only an electric motor would give efficient and desirable service. Steam may be required for other purposes around the plant, which would bring the steam-driven plant into favorable consideration.

The reliability of oil engines as prime movers may be increased by employing units that have 25 to 33 1-3 per cent additional capacity above the actual requirements of the plants. Working at these lower percentages of full-load capacities is not so detrimental, since the fuel consumption, especially that of the Diesel type, is increased but slightly.

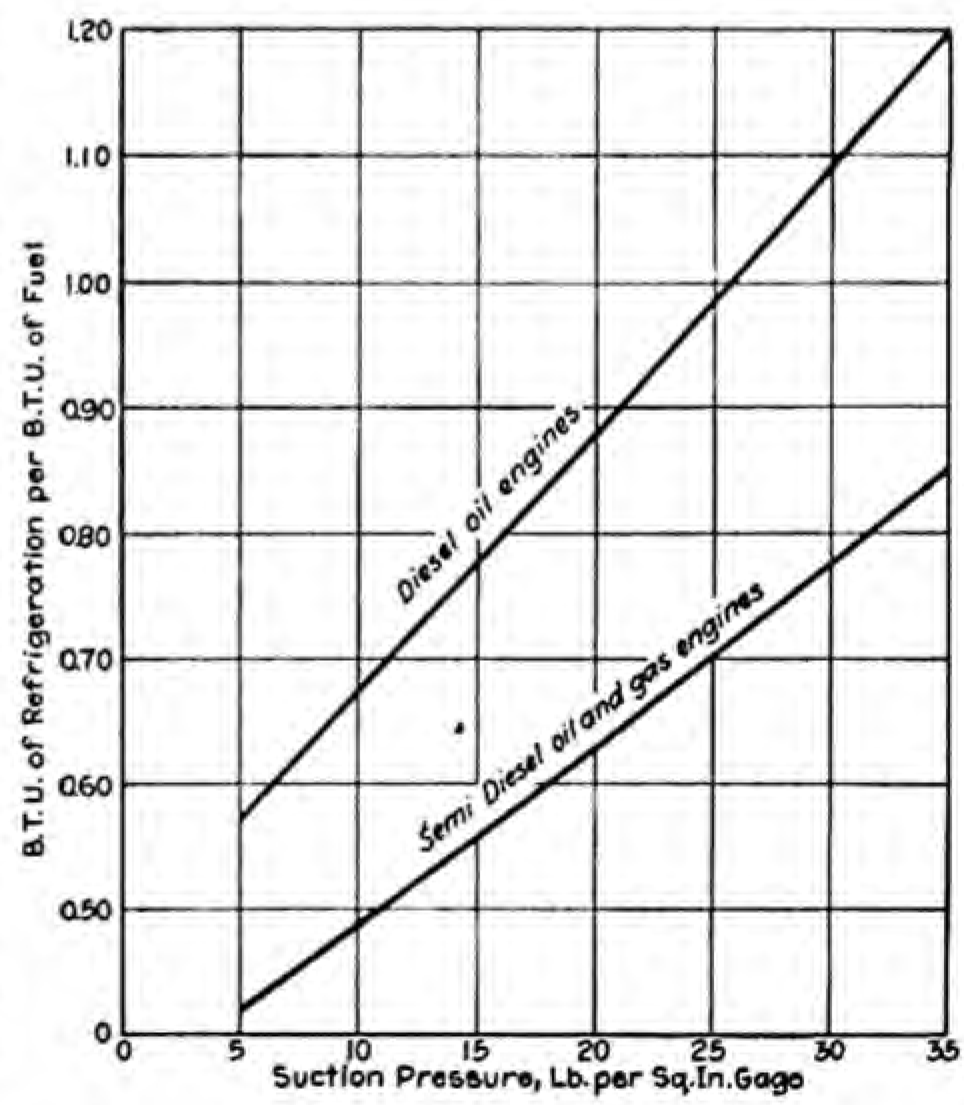

Fig. 211.—Efficiencies of Oil Engine Plants.

In plants where small engines are used, it is a good policy to use a good grade of fuel oil in order to eliminate as much as possible the bad effects that would be produced by the employment of a low-grade fuel. However, in the larger plants, where greater quantities of oil are consumed, it is advisable to use a lower-grade oil in order to reduce the cost of fuel.

Fuel Consumption.—The consumption of fuel oil by the Diesel engine of medium size is shown by Fig. 210 for various percentages of full-load capacity, as is the fuel consumption of the semi-Diesel type. These are the amounts that are guaranteed by the builders of oil engines, although actual consumption is sometimes less.

Relative Efficiencies.—The relative efficiencies of the Diesel and semi-Diesel oil engine plants as a means of producing refrigeration are indicated in Fig. 211. With respect to the amount of refrigeration produced by the expenditure of a Btu. in heating value of fuel, the Diesel-engine driven plant is the most efficient type known.

Relative to the simplicity of operation, the semi-Diesel engines require only an intelligent attendant to keep them in good condition. He should, of course, understand the function of the engines thoroughly. The Diesel type requires an operator that is not only intelligent but also skilled. However, the Diesel type is not too complicated to be thoroughly understood and efficiently handled by the average plant operator, particularly if an effort is made to develop intelligent and skilled operating men in the plants.

QUESTIONS ON CHAPTER XXI.

1. Name the general factors which affect the selection of a suitable prime mover for an ice making or refrigerating plant.
2. What are some local conditions which determine the type of prime mover to be used?
3. How may the steam consumption of steam engines be lowered?
4. At 15 lb. gage suction, how much difference is there in refrigeration output per ton of coal between least and most efficient steam engines listed in this chapter?
5. Why are electric motors being used extensively for driving ice making and refrigerating plants?
6. Discuss the reliability and flexibility of electric motors for driving refrigerating plants.
7. Describe the leading characteristics of synchronous motors.
8. Name some general considerations in reference to the use of oil engines as prime movers.

CHAPTER XXII.

ECONOMICS OF REFRIGERATION.

General Considerations.—Before going into the consideration of the factors which affect the economy or economic efficiency of the ice making and refrigerating plants, it is well to note the definitions of some of the general terms. It seems that the terms used in the discussion of economics are not always used in the same sense by all authorities. It is especially true in the discussion of engineering economics to understand thoroughly the meaning of all definitions and terms.

Generally speaking, economy may be defined as the judicious expenditure of labor, materials, and energy in order to attain a required end or purpose. Economics, therefore, may be defined as the science of the general principles which are to be applied in order to secure maximum economy. Economic efficiency is the ratio of the actual performance to an ideal, or standard performance. This is sometimes termed simply the efficiency of the plant or system.

Engineering may be defined as the systematic application of science to the economic production of commodities. Refrigeration engineering is the application of the basic principles of science to thé economic production of refrigeration. Refrigeration engineering economics, therefore, pertain to that part of economics which is or should be applied in the production of refrigeration.

It should be noted that the province of the refrigeration engineer is now primarily the production of the maximum amount of refrigeration with the minimum expenditure of labor, materials and energy. As the ice making and refrigerating industry grows older, the refrigeration engineer is not so much concerned in the development and perfection of basic refrigeration systems, but he is more vitally than ever concerned with the production of refrigeration with the maximum economy. To the refrigeration engineer, the required end of maximum economy is usually the attainment of the minimum unit cost; but an engineer is often called upon to apply scientific principles and methods, in order to attain the minimum unit costs, and hence, a maximum annual profit.

Fundamentally, economics are concerned with the science of value, price, cost and profit. Value is the exchangeable worth of property or service; it is the intrinsic worth of property or service. Price is the amount of money exchanged for property or service. Unit price is the price per unit of property or service; for example, $10 per ton of ice, or 2c per kw-hr. Cost is the money outlay and the debits incurred in securing a given property or service. It should include not only the price paid by the owner at the time a thing is acquired, but should include also the debits which are chargeable to the thing. This would then include such items as the value of the owner's time, development costs, cost of establishment of business, etc. Profit is the excess of the selling price over cost. It will be observed that the true profit will depend upon the true and actual cost of a thing.

Many business enterprises have been failures; many men have been not only deceived by others, but self-deceived as to the true costs and profits. Many of these failures may be attributed to the lack of knowledge of accurate cost keeping and estimating. It is evident that a small amount of study and effort along the lines of engineering economics will reduce materially the number of such industrial failures.

Cost of Ice or Refrigeration.—In order to determine the true cost of ice or refrigeration, consideration must be given to a number of factors. The actual cost will depend upon the greographical location of the plant, cost of fuel, labor and supplies, relative size of the equipment, the design and efficiency of the different parts of the mechanical equipment, management and administration, etc. Without giving this subject much thought, it would seem readily apparent that the unit cost per ton of refrigeration or per ton of ice could be used as a unit of comparison for plants of different sizes, used for different purposes. On the other hand, when one observes that there are so many factors entering into the costs, it is evident that in order to make an intelligent comparison between plants of different sizes, operating under different conditions, one must not only know the relative unit cost, but also the relative magnitude of the different factors which determine unit costs.

Since ice making is probably the largest commercial application of refrigeration, attention will be especially directed to the cost of ice. The most important factors which combine to determine the actual cost of ice may be tabulated as follows:

Overhead and Administration......	Taxes Insurance Interest Salaries Legal Matters Miscellaneous

Sales and Delivery Expense........	Office Expense Collecting and Soliciting Expense Platform Expense Wholesale and Retail Delivery Expenses Advertising Miscellaneous
Manufacturing Expense	Power Labor Ammonia Oils Supplies Depreciation Repairs Taxes Insurance Incidentals

In addition, consideration must be given to the actual operating conditions in the plant, the relative size of the plant, and the different parts of the equipment, the load factors, the variation in the load, etc. In consideration of the foregoing factors, it is evident that each plant presents an individual problem in itself, so that it is practically impossible to make actual comparisons between the real cost of ice and refrigeration under such variable conditions.

Probably the best method of stating the relative economy or efficiency of an ice making or a refrigerating plant would be to give the ratio of the number of heat units extracted in the refrigeration to the number of heat units expended by the prime mover. This would reduce the comparison to a consideration of relative efficiencies, rather than the relative magnitudes of unit costs. Notwithstanding this fact, however, it is probably advisable to give attention to the various items that enter into the manufacturing cost of ice refrigeration.

The practical engineer is primarily interested in the manufacturing cost or the expense of the mechanical production of ice or refrigeration, and is not particularly concerned with the expenses of the offices, sales and delivery departments. Although some of the items which enter into the manufacturing cost of the ice may be beyond the control of the practical engineer, he should be able to make such records and such calculations which would indicate to himself, as well as to the management of the plant, something in respect to the relative efficiency of operation and the cost of production of ice or refrigeration. The practical engineer, by the use of his plant records and elementary knowledge of bookkeeping and his calculations, will not only be able to determine the relative manufacturing unit cost, but will also be led to make analyses which will show where improvements may be accomplished.

Manufacturing Expense.—As previously indicated, the practical engineer is concerned with the manufacturing expense, or the cost of production of ice or refrigeration mechanically. The items which enter into the cost of production may be divided into two classes, namely, fixed charges and variable or operating charges, the fixed charges being independent of the relative output of the plant and the variable charges depending upon the output or operating costs. These charges may be classified as follows:

Fixed Charges	Depreciation Repairs Taxes Insurance Incidentals
Operating Charges	Power Labor Ammonia Oil Supplies

The fixed charges are constant throughout the year, irrespective of the relative load on the plant. This is due to the fact that the depreciation, maintenance, taxes, insurance, incidentals, etc., go on throughout the year, whether or not the plant is in operation. The variable or operating cost will vary in proportion to the relative time that the plant is in operation.

Fixed Charges.—The fixed charges depending upon depreciation, insurance, taxes, etc., will vary considerably with different sizes of plants, operating conditions, etc. The operating engineer is able to retard considerably the natural depreciation of the apparatus by keeping it in as nearly perfect mechanical condition as possible. By keeping the apparatus in perfect mechanical condition, the expense for repairs and maintenance will be likewise materially reduced. It is evident that the plant should earn enough so that sufficient capital may be set aside each year as a fund to offset the depreciation of the apparatus, so that when the apparatus reaches the end of its useful life, new and up-to-date equipment may be installed. The various amounts charged for the foregoing items are generally stated in terms of the initial investment. The relative magnitude of these amounts and their variations are shown by the following tabulation:

Depreciation	5%— 6%
Repairs	3%— 5%
Insurance Taxes Incidentals	2%— 4%
Total	10%—15%

It will be noted that the total amount of fixed charges will vary between 10 and 15 per cent of the original investment. In all of the calculations which follow, the fixed charge of 15 per cent has been used for purposes of comparison.

It is well to observe the relation between the fixed charge per unit of production and the load factor. The yearly load factor, or simply the load factor, of the plant is the ratio of the actual yearly output to the rated yearly output. This is simply the ratio between the total output of the plant if operated throughout the year at its rated capacity or 365 days of 24 hours or 8,760 hours per year, and the actual output of the plant for the same period, namely, a year.

For example, if the plant is equipped with apparatus for producing one hundred tons of ice per day, the rated capacity of the plant would be 36,500 tons of ice per year. But if the plant is operated only 200 days per year, the actual output would be 20,000 tons per year. The yearly load factor would, therefore, be 20,000 divided by 36,500, which equals 54.8 per cent. If this 100-ton ice making plant was assumed to cost $100,000, the yearly fixed charges may be assumed to amount to 15 per cent. The total fixed charge for one year would be 100,000x0.15 which equals $15,000.

In the foregoing example, when the plant operates at its rated capacity throughout the year, the resulting fixed charge per ton of ice would be equal to $15,000 divided by 36,500 which equals $0.41; but when the plant is only operated 200 days per year, the resulting fixed charge per ton of ice would be 15,000 divided by 20,000 which equals $0.75. From the foregoing, it will be observed that when the load factor is high, the greatest amount of ice will be produced per year, thereby decreasing the fixed charge per ton of ice.

One of the commonly used methods of determining the amount of money to be set aside to offset the depreciation of the apparatus is that known as the straight line method. This method is based on the assumption that if the total investment of the plant, less the salvage or scrap value of the plant, is divided by the weighted life of all parts of the plant, the resulting quotient will be the amount which should be allowed each year to offset depreciation. This is one of the more simple methods which have been devised for this purpose, and eliminates calculations concerning compound interest. The various factors which may be determined by this (after G. F. Gebhardt) method are tabulated as follows:

$$D = \frac{C-S}{n};\ V = (C-S)\left(1-\frac{m}{n}\right)$$

$$d = 100\frac{D}{C};\ A = C - V$$

where D = total accrued depreciation per year
C = total original cost
S = scrap or junk value of plant
n = assumed useful life in years
V = present value
m = age of plant in years
d = rate of depreciation in comparison of total original cost
A = total accrued depreciation

Example—A 15 in. x 30 in. ammonia compressor has been in operation eight years and the first cost originally was $4,000. If the scrap value of the compressor is assumed to be $600 and the useful life is taken to be 20 years, the actual depreciation charge, the actual rate of depreciation, the present value and the total accrued depreciation may be calculated as follows:

Annual Depreciation Charge:

$$D = \frac{C - S}{n} = \frac{4000 - 600}{20} = \$170$$

Depreciation Rate:

$$d = 100\frac{D}{C} = 100\frac{170}{4000} = 4.25\%$$

Present Value:

$$V = (C - S)\left(1 - \frac{m}{n}\right) = (4000 - 600)\left(1 - \frac{8}{20}\right) = \$2{,}040$$

Total Accrued Depreciation:

$$A = (C - V) = 4000 - 2040 = \$1{,}960$$

In using the straight line law for the determination of depreciation charges, it must be borne in mind that the original cost should include the total actual cost of the plant, consideration being given to labor, material, overhead, engineering, interest, etc. The probable, useful life of the apparatus is, of course, a theoretical quantity and must be taken to be the assumed weighted value for all the different parts of the plant.

Coal Consumption and Cost.—The relative amount of coal consumed per ton of ice or refrigeration will depend upon the size of the plant and the relative efficiencies of the apparatus. In the smaller ice making plants from 10 to 100 tons capacity per day, the slow speed Corliss steam engine is generally used as a prime mover. In the largest plants from 100 to 500 tons capacity, such refinements as compound condensing engines, automatic stokers and evaporators are introduced.

For the purpose of making comparisons, the relative average amount of ice produced per ton of coal will be taken from Table 101 for the different sizes of plants. From this table it will be noted that

the smaller plants will produce only from 5 to 6 tons of ice per ton of coal, while the larger plants will produce from 9 to10 tons of ice per ton of coal. The relative consumption of steam per i. hp. per hr. of steam engines working at full rated capacity are shown by the following tabulation:

TABLE 99.—STEAM CONSUMPTION OF ENGINES.

I.h.p.	Simple slow speed non-condensing	Compound slow speed-condensing
100	27.0	20.0
150	26.3	19.5
200	25.7	19.0
250	25.3	18.5
300	24.8	18.1
400	24.1	17.3
500	23.7	16.5
600	23.4	15.8
700	23.2	15.3
800	23.0	15.0
900	22.9	14.7
1,000	22.8	14.5

The relative mechanical efficiency of reciprocating steam engines at full load capacity is shown by the following tabulation:

TABLE 100.—MECHANICAL EFFICIENCIES OF ENGINES.

I.h.p.	Mechanical efficiency per cent
5	80.0
25	83.5
50	85.0
200	86.5
400	89.0
500	90 0
1,000	90.8

The economies shown by Table 101 in reference to the number of tons of ice per ton of coal are based upon average efficiencies of mechanical equipment, operating under average conditions and will be used only for comparisons, and should not be taken to represent the actual figure for any given plant.

The price which must be paid for coal will depend upon the geographical location of the plant and the quality of the coal desired. For purposes of illustration a price of $7.00 per ton is assumed in calculating daily operating costs.

Of course, when the actual cost of coal for a given plant is known, the correct figure should be used, instead of the assumed value of $7.00.

TABLE 101.—ASSUMED INVESTMENTS, FIXED CHARGES, FUEL CONSUMPTIONS, LABOR COSTS, AND MISCELLANEOUS EXPENSE FOR ICE MAKING PLANTS.

Daily Capacity, Tons of Ice	10	20	40	60	100	200	300	400	500
STEAM ENGINE DISTILLED WATER ICE PLANTS									
Investment, excluding land	14,400	21,600	42,300	69,600	120,000	222,000	318,000	408,000	492,000
Fixed charges, 15%	2,160	3,240	6,345	10,440	18,000	33,300	47,700	61,200	73,800
Tons of ice per ton of coal	5	5.6	6.6	7.2	8	9	9.5	9.8	10.00
Total labor expense per day	18.00	22.00	26.00	33.00	43.00	65.00	84.00	102.00	117.00
Ammonia, oil, waste, supplies	2.40	4.80	8.40	9.60	15.60	22.80	28.80	37.20	43.20
ELECTRIC MOTOR RAW WATER ICE PLANT.									
Investment, excluding land	12,600	19,200	39,000	60,600	104,000	192,000	278,000	354,000	420,000
Fixed charges, 15%	1,890	2,880	5,850	9,090	15,600	28,800	41,700	53,100	63,000
Kw.-hrs. per ton of ice	60	59	57	55	53	50	49	48.5	48
Total labor expense per day	14.00	18.00	22.00	29.00	37.00	58.00	75.00	92.00	106.00
Ammonia, oil, waste, supplies	2.40	4.80	8.40	9.60	15.60	22.80	28.80	37.20	43.20
OIL ENGINE RAW WATER ICE PLANTS.									
Investment, excluding land	13,800	21,000	46,200	73,200	126,000	237,000	342,000	438,000	534,000
Fixed charges, 15%	2,070	3,150	6,930	10,980	18,900	35,550	51,300	65,700	80,100
Gallons of fuel oil per tons of ice	10	7.7	5.4	4.8	4.4	4.3	4.2	4.1	4.00
Total labor expense per day	14.00	18.00	22.00	29.00	37.00	58.00	75.00	92.00	106.00
Ammonia, oil, waste, supplies	2.40	4.80	8.40	9.60	15.60	22.80	28.80	37.20	43.20

Electric Power Consumption and Cost.—The relative amount of current consumed in driving machinery in ice making and refrigerating plants will depend upon the size of the plant and the design and efficiency of the motor. In smaller plants requiring horsepower up to 150, the polyphase induction motor is usually employed for this purpose. When the power requirements are above 150, it is customary to use the synchronous motor.

The current consumption of electric motors is not subject to so large a variation as the consumption of coal by steam-driven plants, due to the fact that the efficiency of the different-sized motors is more nearly equal. The relative amounts of electric current consumption in kilowatts per ton of ice for ice making plants is shown by Table 101 for comparison. These are yearly averages and may appear somewhat high at first thought. Sometimes claims are made for much lower current consumption per ton of ice, but in this case it is generally an average for a shorter period of time than a year.

The relative speed and efficiency of standard induction motors are shown by the following tabulation:

TABLE 102.—EFFICIENCY OF ELECTRIC MOTORS.

Horsepower	Full load speed at 60 cycles	Efficiency in per cent of full load
2	1120	84
5	1120	86
10	1135	86
20	1135	87.5
50	850	88
200	575	92

The cost of current for electrically operated ice making and refrigerating plants will depend upon the geographical location, the relative size of the plant, the type of motors, etc.

Fuel Oil Consumption and Cost.—The comparative consumption of oil in gallons of fuel oil per ton of ice is shown by Table 101. These values are based upon average conditions and will depend upon the relative size of the plant and the oil engine. The oil engine has an inherent advantage of high thermal efficiency, which accounts for the relatively low fuel consumption.

The cost of 5 cents per gallon of fuel oil has been assumed for the calculations which are to follow.

Daily Operating Expenses.—As previously indicated, the assumed fuel and the power requirements for different sized ice plants are shown by Table 101. In addition to the fuel and power expense, the cost of labor, ammonia, oils, supplies, etc., will also enter into the daily operating expense. In order to study the various factors which enter into the total manufacturing cost of ice, the value of the labor and

miscellaneous expenses as shown by Table 101 have been assumed. These values may be taken to be the average, and will vary with each individual plant. The amounts of labor required for different kinds of ice making plants does not seem to vary materially, with the possible exception that the steam engine driven distilled water ice plant will require somewhat more labor for operation.

The labor expenses for the electrically and oil engine driven raw water ice plant have been assumed to be the same. Assuming that the power and fuel costs are 1¼c per kw.-hour, $7 per ton for coal, and 5c per gallon for fuel oil, the daily fuel and power expenses have been calculated according to the consumptions indicated by Table 101. Those total fuel costs are shown in Table 103. The total daily operating expense, consisting of labor, fuel, oil, power and sundries, is shown by Table 103, also for different sizes of ice making plants, using different forms of prime movers.

Investment Expense.—In order to study how the fixed charges on the investment in ice making plants will affect the total manufacturing cost per ton of ice, the relative investment required for different-sized plants must be assumed. This is shown by Table 101. It must be remembered that these costs of plants will depend upon a great number of factors, and will be individual with each plant. Those shown by Table 101 are assumed simply as a matter of comparison. The investment charges shown by this table include the mechanical equipment and the building, but do not ordinarily include the value of the land, which will vary considerably with the location. Table 101 also shows the correponding fixed charges when these are taken to be 15 per cent of the total investment in the building and mechanical equipment of an ice plant.

Load Factors.—The yearly load factors must be taken into consideration in order to arrive at an accurate estimate of the true costs of manufacturing ice or refrigeration. The yearly load factor is simply the ratio of the number of days during which the plant is operated at full capacity, to the number of days in the year, namely, 365. Ice making and refrigerating plants will ordinarily operate from seven to eleven months per year. This would mean that the plant would be closed down from one to five months, depending upon the relative size of the plant, the demand for ice, the use of an ice storage house, etc. The number of months of operation per year, the number of days' operation per year, and the corresponding yearly load factors are shown by the following tabulation:

Months Operation per year	Days Operation per year	Days Shut Down per year	Full Load Factor
11	335	30	91.8%
9	274	91	73.1%
7	213	152	58.4%

TABLE 103.—DAILY OPERATING COSTS, FOR ICE PLANTS, FOR ASSUMED CONDITIONS.

Daily Capacity, Tons of Ice	10	20	40	60	100	200	300	400	500
STEAM ENGINE DISTILLED WATER ICE PLANTS.									
Labor	18.00	22.00	26.00	33.00	43.00	65.00	84.00	102.00	117.00
Coal at $7.00 per ton	14.00	25.00	42.40	58.30	87.50	155.50	221.00	288.00	350.00
Sundries	2.40	4.80	8.40	9.60	15.60	22.80	28.80	37.20	43.20
Net expense	34.40	51.80	76.80	100.90	146.10	243.30	333.80	427.20	510.20
ELECTRIC MOTOR RAW WATER ICE PLANTS									
Labor	14.00	18.00	22.00	29.00	37.00	58.00	75.00	92.00	106.00
Current at 1¼c per kw.-hr.	7.50	14.75	28.50	41.25	66.30	125.00	180.00	242.20	300.00
Sundries	2.40	4.80	8.40	9.60	15.60	22.80	28.80	37.20	43.20
Net expense	23.90	37.55	58.90	79.85	118.90	205.80	283.80	371.40	449.20
OIL ENGINE RAW WATER ICE PLANTS.									
Labor	14.00	18.00	22.00	29.00	37.00	58.00	75.00	92.00	106.00
Fuel oil at 5c per gallon	5.00	7.70	10.80	14.40	22.00	43.00	63.00	82.00	100.00
Sundries	2.40	4.80	8.40	9.60	15.60	22.80	28.80	37.20	43.20
Net expense	21.40	30.50	41.20	53.00	74.60	123.80	166.80	211.20	249.20

It is further assumed that the plant operates at full load capacity during the period of operation.

Comparative Manufacturing Costs.—As previously indicated, the total manufacturing cost is made up of two parts—first, the fixed charges, which consist of depreciation, repairs, taxes, insurance and incidentals; second, the operating charge, which consist of power, labor, ammonia, oil and supply expenses. It was also indicated that the cost must be reduced to a yearly basis in order to arrive at accurate results. In order to study the relationship between these various costs it is well to calculate them for a given plant. For this purpose an electrically driven raw water ice making plant, operating seven months at full capacity, and having a capacity of 100 tons of ice per day, will be used for illustration. The seven months' operation at full capacity would be equivalent to 213 days, or a yearly load factor of 58.4 per cent. The investment for the plant, excluding the value of the land, may be assumed to be \$104,000. Assuming that the fixed charges amount to 15 per cent of this investment, the corresponding expense for these fixed charges would be \$104,000 $\times$ 0.15, which equals \$15,600. The cost of labor required for a plant of this size may be taken from Table 101 and will be \$37.00 per day. The cost of the current for the plant at 53 kw.-hr. per ton of ice and at 1¼c per kw. would be 100 $\times$ 53 $\times$ 0.0125 = 66.30. The expense for sundry items, such as ammonia, oil, waste and supplies, may be taken from Table 101 and will be \$15.60. The total daily expense would, therefore, be \$37.00 plus \$66.30 plus \$15.60, which equals \$118.90 per day.

The total operating expenses of the plant for a period of operation of 213 days would be equal to 213 $\times$ \$118.90, which equals \$25,325.70. Some expense for labor will be incurred to overhaul and repair the plant in order to keep it in perfect mechanical condition during the remainder of the year. For this purpose it will be assumed that the total labor expense will be incurred during the remainder of the year. This would amount to 152 $\times$ \$37.00 = \$5,624. The total yearly expense would, therefore, be made up of the operating expense for 213 days, the labor expense for 152 days, and the fixed charges. This would be equal to \$25,325.70 plus \$5,624 plus \$15,600, which equals \$46,549.70. Operating the plant at full capacity for 213 days, the total number of tons of ice manufactured per year would be 213 $\times$ 100 which equals 21,300. The corresponding manufacturing cost per ton of ice would, therefore, be \$46,549.70 diveded by 21,300, which equals \$2.18 per ton of ice produced per year. All of the foregoing values are shown in the following tabulation:

TABLE 104.—A 100-TON ELECTRIC MOTOR RAW WATER ICE PLANT.

Seven months' full operation = 213 days = 58.4% load factor	
Investment, excluding land	$104,000.00
DAILY OPERATING EXPENSE	
Labor, cost per day	$ 37.00
Current cost per day, 100 × 53 × 0.0125	66.30
Ammonia, oil, waste and supplies	15.60
Net daily expense	$ 118.90
YEARLY OPERATING EXPENSE	
Operating expense for 213 days (213 × $118.90)	$ 25,325.70
Labor expense for 152 days (152 × $37)	5,624.00
Fixed charges at 15% ($104,000 × 0.15)	15,600.00
Total yearly expense	$ 46,549.70
Tons of ice manufactured per year, 213 × 100	$ 21,300.00
Manufacturing cost per ton, $46,549.70 ÷ 21,300	2.18

In the same manner the cost per ton of ice has been calculated for electric drive plants on the basis of data in Table 101 at various annual load factors.

The data thus obtained has been plotted in two curves to show how load factor and plant size affects unit cost. Fig. 212 is a comparison of the costs in a small and a large plant when the load factor varies between 50 and 100 percent. The relative slope of these curves indicates the relative rise of the cost of production of ice as the load factor is reduced. Another interesting consideration that may be derived from the calculation is the effect of the relative size of the plant upon the cost of production of ice. For illustrating this point, the electrically driven raw water ice plant, operating at a load factor of 58.4 per cent, has been selected. The corresponding costs per ton of ice have been taken from the calculation based on Table 101 and plotted on Fig. 213. A smooth curve has been drawn through these points. The rise of this curve shows how rapidly the cost of production of ice increases as the size of the plant decreases.

Unit Operating Costs.—Overall operating costs are subject to many variables as shown before. Real control of costs is best obtained by breaking down any lump sum into the various units and considering each of them on the basis of the plant design, load factor and the other particular conditions existing in the plant. For instance if we take an ice plant as an example it probably would be helpful in studying costs to allocate manufacturing labor to the following divisions:

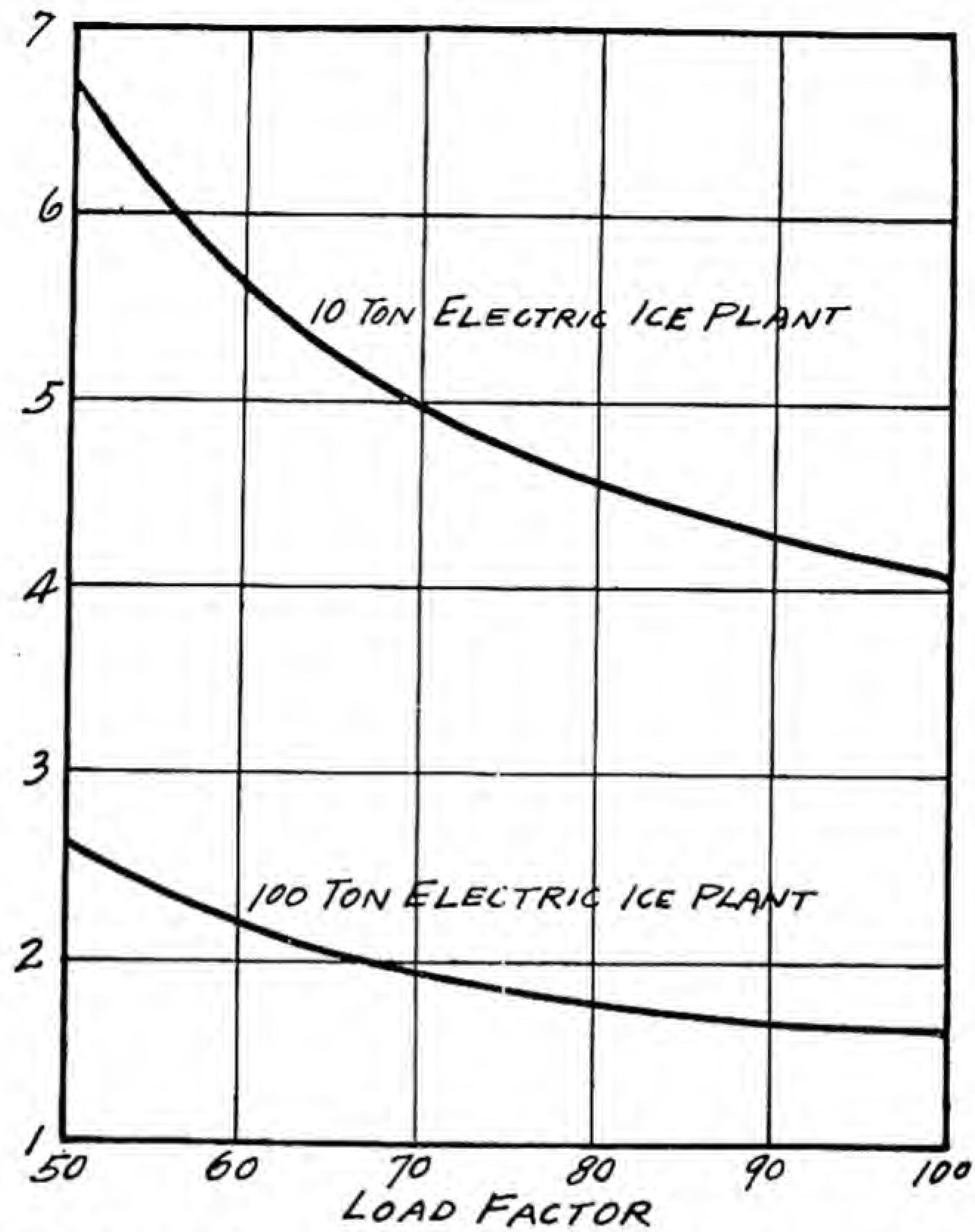

Fig. 212.—Effect of Load Factor on Cost Per Ton of Ice.

Engineers	Ice pullers
Firemen	Air man
Oilers	Crane man
Coal passers	Unloading coal
Ash handlers	Ante-room
Boiler cleaners	Cleaning up

Handling labor consist of the following items:

Loading cars	In and out storage
Helping teamsters load	Platform man

Repair labor consist of the following items:

Ice cans	Crane
Ice tanks	Cooling system
Pumps	Water supply system
Boilers	Condensers
Compressors	Hoists

Buildings

The accounting methods in various plants will not be uniform but some such method will be helpful in cost keeping and studies. Of course the total figures on the various divisions of costs are helpful in charting and studying the general trends of plant operating costs. For production labor they may be set up in the following fashion:

Total manufacturing labor	Total repair labor
Total handling labor	Total payroll

Supplies used in production are divided as follows:

Ammonia	Boiled fuel
Oil	Water
Electric energy	Other supplies

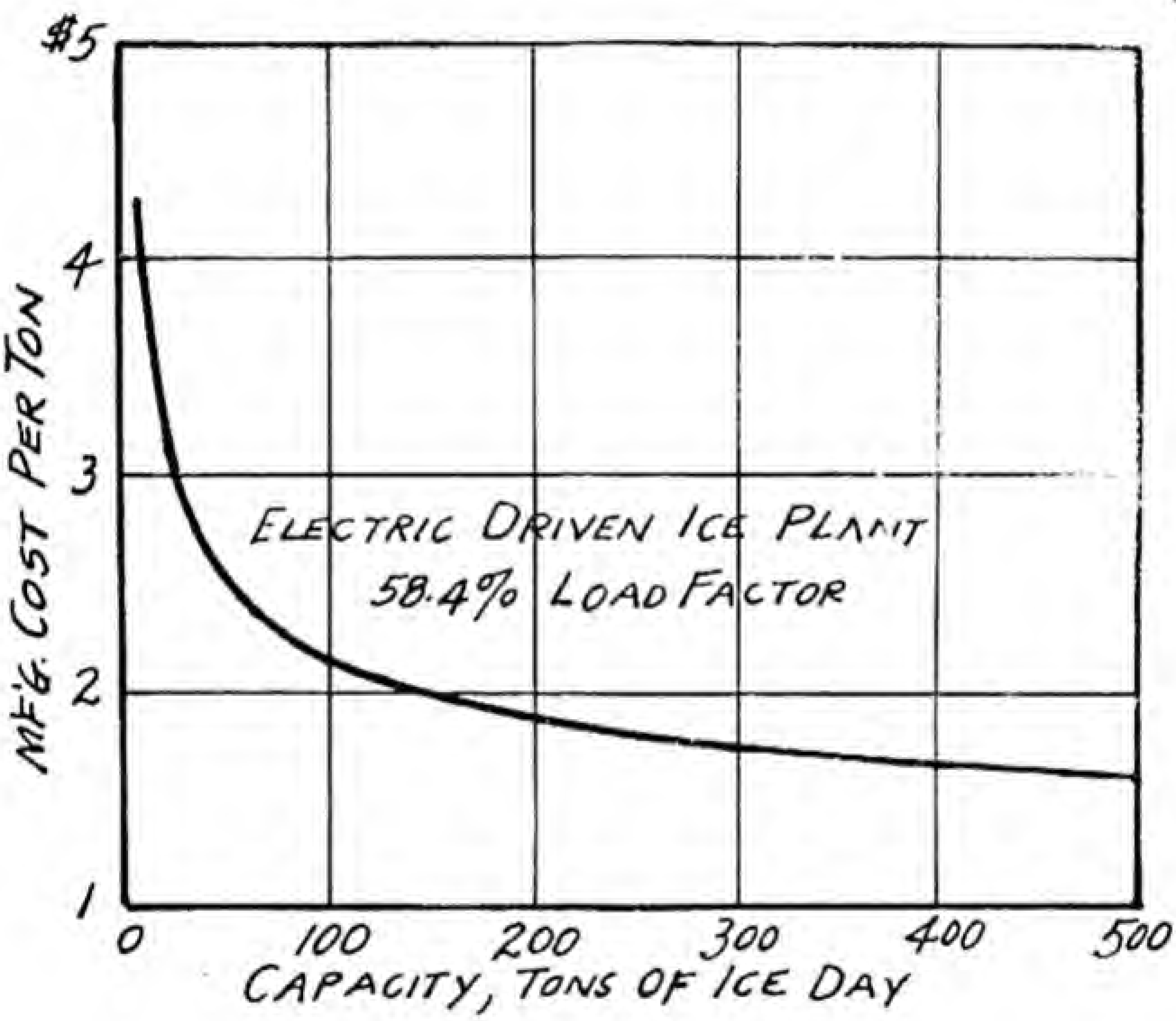

Fig. 213.—Effect of Size of Plant.

Unit costs are more directly controllable by the engineer than are overall costs. For instance he can do everything possible to maintain the refrigerating system in tight condition and thus reduce the cost of refrigerant per unit of output. By careful operation he can also keep the fuel or power cost per ton refrigeration at a minimum for the particular plant.

Overall costs however, may not be completely in his control. For example the plant design and equipment may be old and it may not be possible to go below a certain figure for annual fuel cost. Perhaps such

a situation exists where old steam boilers and engines are the source of power and electric drive or internal combustion engines would reduce fuel, power or labor costs. The decision in regard to replacement of present equipment will probably be the function of someone other than the engineer. He can, however, keep present costs accurately and make careful analyses of savings to be made by modernization so that management will have proper facts on which to base its decisions.

QUESTIONS ON CHAPTER XXII.

1. Name three different classes of expenses entering into the cost of ice and describe some of the important items in each classification.

2. What are the two general classes of expenses entering into the manufacturing cost of ice or refrigeration? Name some important items in each class.

3. How may the power consumption costs be reduced?

4. An 18-in. by 36-in. compressor originally cost $8,000, has a scrap value of $800, has been in operation twelve years, and has an expected life of twenty years. Find the annual depreciation, depreciation rate, present value, and total accrued depreciation.

5. What is meant by load factor?

6. How does the load factor affect the manufacturing of ice or refrigeration?

7. How may depreciation of apparatus and machinery be retarded?

8. What is the estimated yearly expense in a 60-ton raw water ice plant, costing $70,000, which operates 150 days per year?

9. What would be the yearly estimated expense if the foregoing plant operates 335 days per year?

10. What would be the estimated manufacturing cost per ton of ice per year in problems 8 and 9?

CHAPTER XXIII.

GENERAL CONSIDERATIONS.

Engine Room Records.—The observation and the recording of the temperatures, pressures, etc., about the ice making or refrigerating plant is one of economic importance. In the first place, it makes it possible to determine more accurately the production cost per ton of ice or per ton of refrigeration. The determination of the magnitude of the production costs makes it possible to compare this cost with the other items that enter into the total manufacturing cost for a ton of ice or refrigeration.

The study of these records also leads to better efficiency of operation. By means of the information contained in the egine room records, the engineer is able to ascertain the operating efficiencies of not only the separate parts of the plant, but also for the plant as a unit.

In the second place, the engine room log establishes a permanent record of all the operating conditions about the plant. In order for the log to represent the true working conditions as nearly as possible, the data should be entered upon the logsheet several times during the day. One of the most important considerations in the securing of accurate records of working conditions is that pertaining to the instruments for indicating the various temperatures, pressures, quantities, etc. It is obvious that the record will be of little value unless such instruments are accurate, or unless their percentage of error is known.

Another important consideration is that of the suitable location of such instruments. These instruments should preferably be located in convenient places for inspection by the operating engineer.

Kind of Engine Room Logs.—It is probably true that each particular plant will have its own individual engine room log. This is due to the fact that although the plants may be used for the same ultimate purpose, the working conditions of the various units of apparatus will vary with the different plants. In addition, the plants will have different numbers and kinds of units of apparatus. In view of this fact, it is probably advisable for the engineer to develop a log suitable for his

own particular plant. One of the important things in designing an engine room log for the practical operating engineer is that the log should not be too complicated, nor should it require frequent reading of the thermometers, gauges, etc. The engine room log should portray, however, all of the working conditions about the plant, after which the chief engineer may make suitable calculations and comparisons from this and other data.

Due to the fact that frequent readings of the various gauges, thermometers, etc., will become monotonous to the operating engineer, and due to the fact that the conditions in the ice making and refrigerating plant are fairly steady, the records in many plants are taken only every four hours. In some ice making plants, additional data would have to be recorded other than that which is shown by the sample log, while in some plants these sample logs contain some columns for which the data would be missing, due to the lack of proper thermometers, gauges, etc. The log sheets should be in a pad form, or suitably mounted in a book or on a record board. Of course, two copies of the log should be made at each entrance of the data, the original copy being sent to the office, while the carbon copy is retained by the chief engineer. A looseleaf notebook provides a desirable method of filing the separate log sheets. When the plant is driven electrically, it is necessary to record only the initial and final electric motor reading for each day. These together give the current consumption during each particular day. The chief engineer of an ice making or refrigerating plant should be made responsible for the correctness of all data entered on the logs.

Uniform Engine Room Logs.—A daily uniform engine room log for ice making and refrigerating plants was presented at the ninth annual meeting of the National Association of Practical Refrigerating Engineers at Oklahoma City, Oklahoma, in December, 1918. After a study of many of the logs existing at that time, the log as shown by Fig. 214 was finally adopted by the convention. In this engine room log, the readings are taken every three hours, and suitable spaces are provided for the recording of the required data for both ice making and refrigerating plants driven by the various kinds of power. To facilitate the work of the comparison of data at the end of every month, a monthly uniform engine log as shown by Fig. 215 was compiled and adopted. On this the average value of the various quantities entered on the daily log are recorded for each day of the month. At the end of the month the average of the daily quantities may be averaged to give the total monthly average.

In addition to the presentation of the data by means of the log, it may also be presented in a graphical manner. These data, when plotted

DAILY UNIFORM ENGINE ROOM LOG ADOPTED BY N. A. P. R. E.

Plant__________ Date__________ 19____

Time	Steam Pressure	Condenser Pressure			Temperature of Discharge Gas			Return Pressure			Temperature Return Vapor			R P M			Water Temperature											Brine Temp		Temperature Freezing Tanks			Temperature Rooms						Temperature Ice Storage	Vacuum Gage	Draft Gage	Outside Temperature	Fuel Used	CO_2
		Machine No			Machine No			Machine No			Machine No			Machine No			To Heater	To Boiler	To Cooling Tower	From Cooling Tower	To Condenser	From Condenser	Supply	From Absorber	From Rectifier	To Precooler	To Cans	To Storage	From Storage	Tank No			Room No											
		1	2	3	1	2	3	1	2	3	1	2	3	1	2	3														1	2	3	1	2	3	4	5	6						
7 A.M.																																												
10 A.M.																																												
1 P.M.																																												
4 P.M.																																												
7 P.M.																																												
10 P.M.																																												
1 A.M.																																												
4 A.M.																																												
Ox																																												

Machine No	Time				
	Start	Stop	Start	Stop	Total
1					
2					
3					

Total No Blocks Pulled, Day	
Total No Blocks Pulled, Night	
Total No Blocks in Ice House	
Total No Blocks in Dump Room	
Total Tons Pulled	
Total Tons on Hand, 7 A.M.	
Quality of Ice	

Meter Readings, 7 A.M.	
Oil	
Gas	
Watt	
Water	
Brine	
Tons of Coal Used	
Cars Shipped	

Engineers	
Firemen	
Tankmen	
Vaultmen	
Extra Labor	
Chief Engineer	

Material Received	Material Used

Remarks (Shut Downs, Accidents Etc.)

Signed__________

Published by Ice and Refrigeration, Chicago – Official Organ

Fig. 214.—Engine Room Log Adopted by N. A. P. R. E.

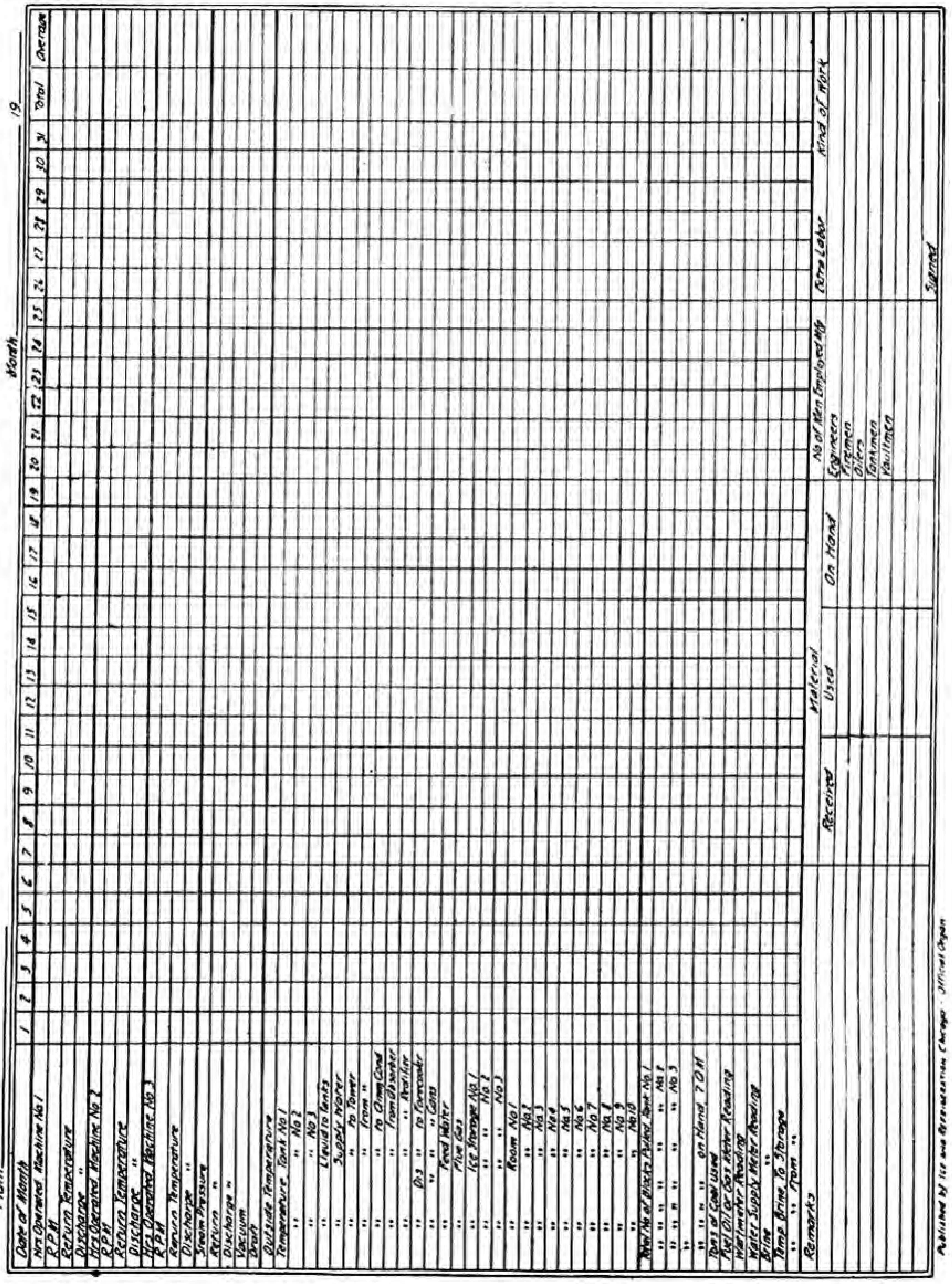

Fig. 215.—Monthly Summary for N. A. P. R. E. Log.

on diagrams having rectilinear coordinates with equally spaced decimal divisions, afford an excellent means for the study of all conditions about the plant. The data from a log may be plotted on charts using the horizontal distances to represent the time of observation, and the vertical distances to represent the various data as recorded. Such charts show at a glance the uniformity of the various data.

Use of Operating Data.*—The intelligent record and use of the physical and operating data from an ice making and refrigerating plant is beneficial from several view points. In the first place, it is impossible for the operating engineer to carry all of the essential facts in his mind, due to the diversfied line of equipment usually installed in such plants, and to the change of operation characteristics which depend upon the capacity factor, season, etc. In the second place, the record serves as a guide for the various shifts of operating engineers. In the third place, it provides a visible record for the chief engineer and superintendent. In the fourth place, it provides a means of comparison of the operating characteristics, costs, etc., from month to month and year to year. In the fifth place, it provides data from which the manufacturing cost may be determined. It is obvious, then, that the proper record and use of such data enhances the efficiency of operation as well as the maintenance of the mechanical equipment installed in ice making and refrigerating plants.

Due to the various types of equipment used in different kinds of plants it is not practical to develop a uniform log that will serve all of the requirements of the individual plants. All of the local conditions must be considered in laying out a suitable log sheet. In addition to the data recorded on the log sheet, the engineer and superintendent should keep an accurate record of machinery specifications and other important physical data, pertaining to each and every part of the equipment. These data are essential in checking up the operation of the various parts of the equipment, as well as useful in ordering of new parts or replacements.

The following tabulation gives some of the things which may be recorded under the heading of "specifications and physical data" for an ice plant:

ACCUMULATOR

Type	Make

AGITATOR.

Type	Make
Size	R.p.m.

* A. J. Authenrieth, N.A.P.R.E. Proceedings, 1927.

Air Blower.

Type	Phase
Motor Size	Capacity
Make	Frequency
R.p.m.	Pressure
R.p.m.	

Air Compressor.

Type	Motor or engine size hp
Make	R.p.m.
Size	Voltage
Cylinder (No.)	Phase
R.p.m.	Frequency
Connected to motor or engine.	

Air Cooler Coils.

No. of stands	Pipes high
Length of stands	Size pipes

Air Dehydrator.

Make	Type
Size	

Air Receiver.

Diameter	Length

Air Washer.

Type	Size
Make	

Ammonia Compressor.

Make	Belted or direct connected
Size (bore and stroke)	Motor or engine size
Number of cylinders	R.p.m.
Single or double acting	Voltage
R.p.m.	Phase
Refrigerating capacity	Frequency

Cans for Freezing Tank.

Total	Inside length
Number hoisted together	Outside length (overall)
Inside bottom dimensions	Nominal weight of ice block
Inside top dimensions	

Condensers (D. P. and Atmos.).

Make	Length of pipes
Type	Sq. ft. of condenser surface
Number stands	Size of liquid connection
Pipes high	Size of gas connection
Size of pipes	

Shell and Tube Condensers

Make	Number of tubes
Type	Length of tubes
Number of shells	Sq. ft. of condenser surface
Diameter of shell	Size of liquid connection
Diameter of tubes	Size of gas connection

Core Sucking Unit.

Make	Motor hp.
Type	Voltage
Size	

Crane.

Make
Capacity (tons)
Capacity (cans ice)
Size motor
Voltage motor

Cooling Tower.

Location
Size
Capacity
Guaranteed performance

Forecooler.

Make
Diameter
Height
Size pipe in coil
Length of pipe in coil
Sq. ft. of cooling surface
Is ammonia expanded through forecooler coil or is returning gas to the compressor used?

Freezing Tank

Number of cans long
Number of cans wide
Total cans
Number of stands of pipe
Number of pipes high
Length of pipe
Size of pipe
Total cooling surface in pipes
Number of coils
High or low-pressure drop tube or fixed tube

Pumps, Condensing Water.

Make
Type
Size
Capacity
Head
Motor hp.
Voltage

Pump, Deep Well.

Make
Type
Size
Capacity
Head
R.p.m.
Motor hp.
Voltage

Purge Drum.

Make
Type
Size
Location
Where connected with system
Does drum contain refrigerating coils?

Scoring Machine.

Make
Type
Size

Storage Room (Day).

Length (inside)
Width
Height
Total area
Area adjacent refrigerating rooms
Storage capacity (tons)
Direct expansion of brine refrigeration
Size pipe
Lineal feet of pipe

Storage Room (Season).

Length
Width
Height
Total sq. ft.
Sq. ft. adjacent to refrigerating rooms
Storage capacity (tons)
Direct expansion of brine
Size pipe
Lineal feet of pipe

SPRAY POND.

Number of nozzles
Make of nozzles
Size nozzles
Size pipe to pond

WATER FILTER.

Make
Type
Size

WATER SOFTENER.

Make
Type
Size

WATER.

Source of supply
Cost of 1000 gal. if purchased
State here any unusual arrangement for supply or disposal of water
Point of entry of make-up water (such as condenser pit, spray pond, etc.)

In addition to the foregoing items it is advisable to record the manufacturer's serial or machine numbers of the various pieces of equipment. Parts lists frequently are useful in ordering new parts for repairs. Drawings and data furnished with new machines and equipment should be properly marked for identification and filed for future use. All these records prove very useful from time to time.

Many units of industrial equipment will have a life of 20 or more years and it is impractical to rely on the memory of individuals for the necessary details about them. Records well preserved and properly indexed therefore provide the only assurance that full knowledge of the data about the plant and its equipment will be available whenever needed.

QUESTIONS ON CHAPTER XXIII.

1. Why is it important to keep a record of the various temperatures, pressures, etc., in ice and refrigerating plants?

2. Upon what does the value of engine records depend?

3. Why is it necessary for nearly every plant to have its own engine room log?

4. Name several important things to be recorded on the engine room log.

5. Describe the N. A. P. R. E. engine room logs.

6. Make up a sample engine room log for a 60-ton raw water ice plant, electrically-driven, which has one ice tank.

7. What is the relationship between the recording of plant operating data and plant operating efficiency?

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